AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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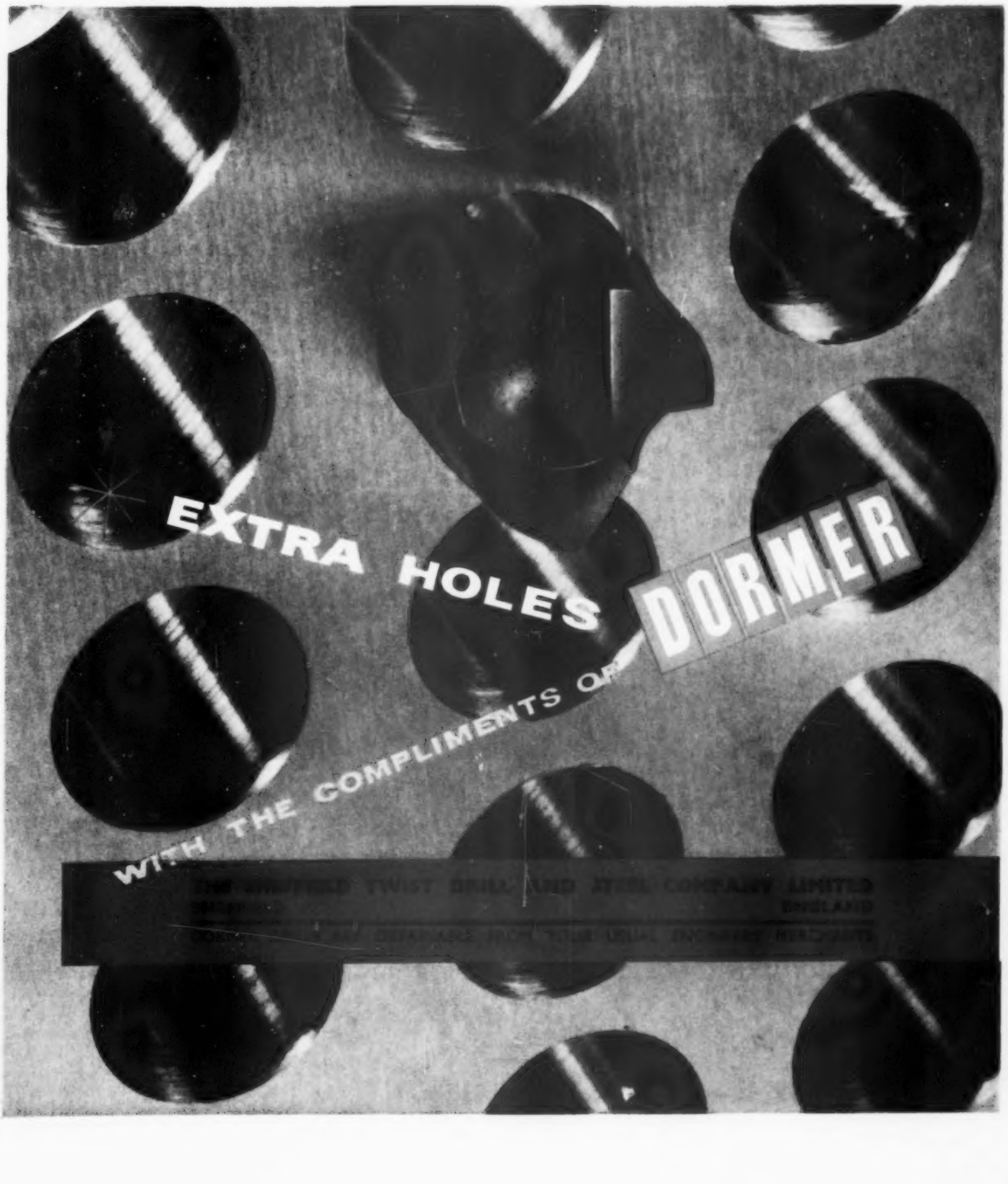

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How BORG & BECK

The artist's impression shows one of the many testing machines used in the evolution and testing of Borg & Beck clutches.

The operator is checking torque capacity—the clutch is mounted on a flywheel; this is revved up to a certain speed and the Lockheed brake (left) is applied, and the torque at which the clutch actually slips is recorded.

'Coefficient' stability of clutch facings is also tested on this machine.

Other machines, some operating at up to 7000 clutch engagements a day, test clutch facings for coefficient of friction, wear and other characteristics.

It is this constant research which maintains the progress and quality of Borg and Beck clutches.

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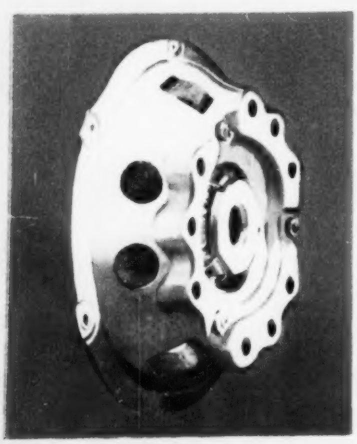
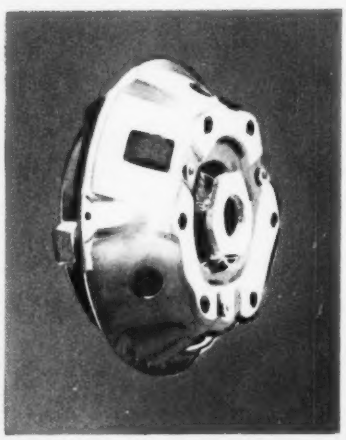
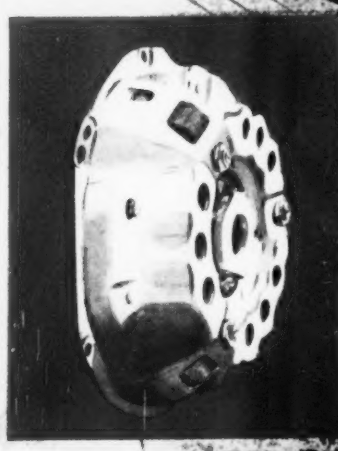
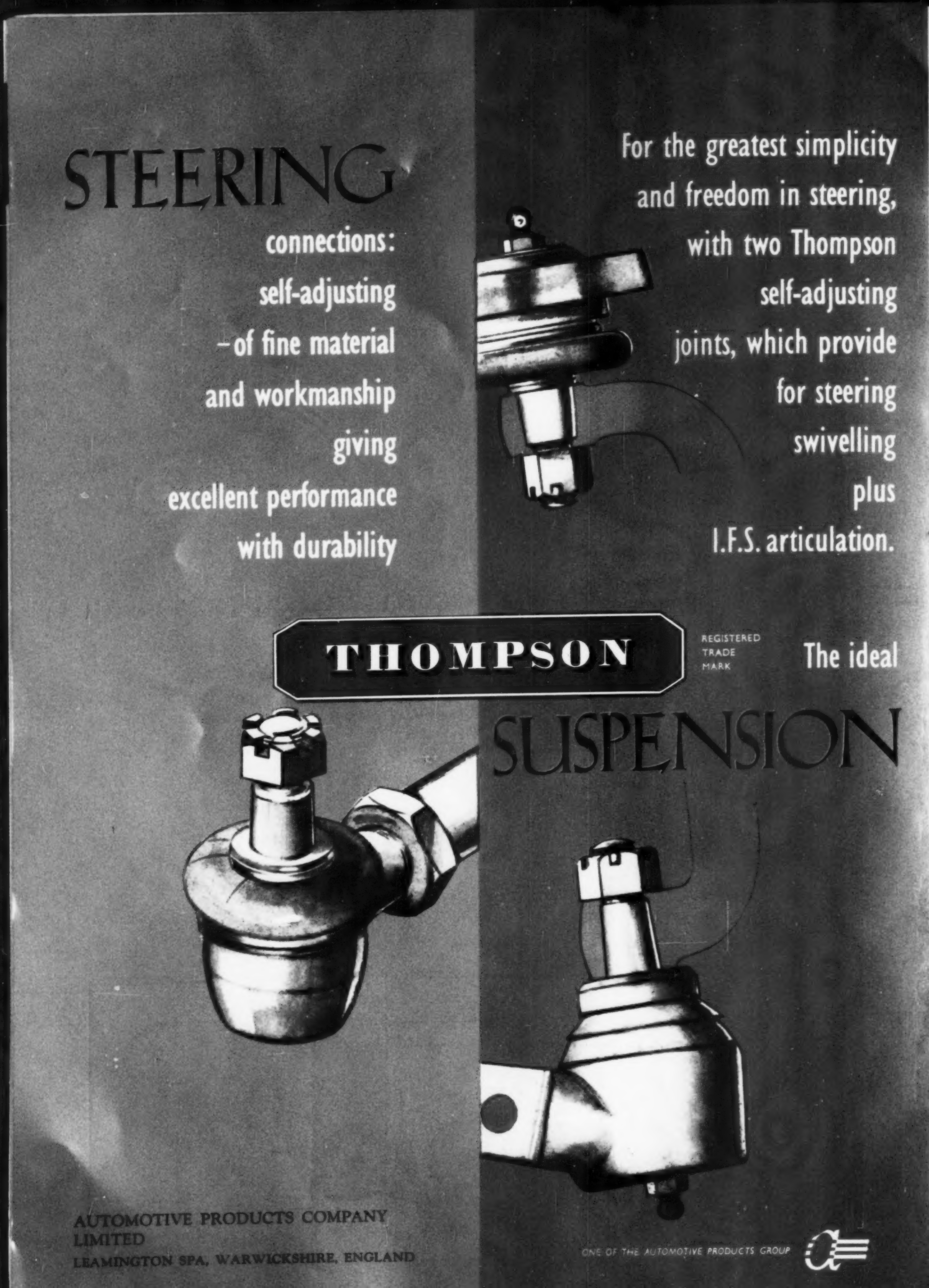







quality is maintained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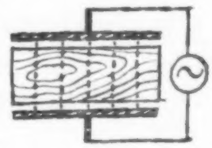
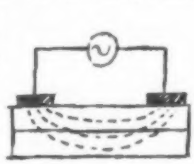
Dielectric Heating - 1

When an electrically non-conducting material is placed between two metal plates, called electrodes, connected to an A.C. supply, the alternating electrostatic field between the electrodes considerably speeds up the molecular movements in the material (termed a 'dielectric') as a result of which the temperature of the material under treatment rises. A similar effect is produced where the two electrodes are positioned on the same side of the dielectric; in this case the electrostatic field between them is generally known as a 'stray' or 'fringe' field.



For industrial application, the applied voltage of the order of 15,000 volts supplied by an electronic generator alternates at frequencies of some millions of cycles per second.



The amount of heat generated in the dielectric is determined by the frequency, the square of the applied voltage, the dimensions of the object and a physical property of the material termed "loss factor" and is represented by the equation:—

$$\text{Power} = 1.41\ E^2\ f.\ F\ \frac{A}{t} \times 10^{-15} \text{ kilowatts.}$$

Where E = applied voltage, f = frequency, F = loss factor.

A = area of the dielectric in square inches.

t = thickness of the dielectric in inches.

F, the loss factor, is itself equal to the expression K Cos Ø in which:

K = the dielectric constant, a measure of the property of the material to retain energy arising from disturbance of its molecular structure.

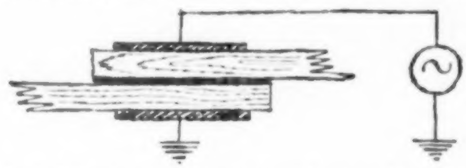
Cos Ø = the dielectric power factor of the load, that is the ratio of the power (in watts) to the product (in volt-amperes) of the voltage and current. This is a characteristic property of the material.

Therefore, "loss factor" is a property of the material and a measure of the ease with which it can be heated by this method. Like other physical properties, it varies considerably for different substances. The equation shows that the heat generated in a dielectric is proportional to its loss factor, but the rate of rise of temperature will also depend upon its specific heat and density. The following table gives approximate values of the dielectric constant, power factor and loss factor of a few typical dielectric materials for frequencies around a million cycles a second.

MATERIAL	DIELECTRIC CONSTANT	POWER FACTOR	LOSS FACTOR
Natural Rubber	2.9	0.02	0.058
Oak, dry	3.3	0.04	0.132
P.V.C.	5.3	0.06	0.318
Urea formaldehyde	7.0	0.03	0.21
'Bakelite' resin	6.0	0.03	0.18
Nylon	3.7	0.05	0.185
Water, pure	80.0	0.03	2.40
Water, tap	80.0	0.5 5.0	40 400

The high loss factor of water means that materials which are difficult to heat when completely dry will often heat efficiently when moisture is present. The voltage must be increased towards the end of the process in some cases to remove the final moisture traces, the reduction in loss factor as the material dries out providing a safeguard against overheating.

Dielectric heating of a homogeneous material is a straightforward application, heat being generated uniformly throughout. If the workpiece is made up of a number of materials, each material will heat up uniformly but each at a rate depending upon its loss factor, thermal properties and density. The degree of temperature uniformity throughout the workpiece will then depend upon the extent to which thermal conductivity can equalise different rates of heating.



Such different rates of heating can be turned to good account in certain applications. For example, in wood glue setting, the glue lines heat up much more rapidly than the wood pieces being joined and the glue sets before the wood heats up substantially, wood having a lower loss factor than glue. Dielectric heating does not depend upon any external heat source to transfer heat by conduction, convection or radiation to the surface of the charge and from thence to the interior by conduction.

Instead, heat is generated within every particle of a body placed in the dielectric field and, depending upon the uniformity of such a body, an even and extremely fast temperature rise can be achieved.

For further information get in touch with your Electricity Board or write direct to the Electrical Development Association, 2 Savoy Hill, London, W.C.2. Telephone: TEMple Bar 9434.

Excellent reference books on electricity and productivity (8/6 each, or 9/- post free) are available—"Induction and Dielectric Heating" is an example.

E.D.A. also have available on free loan in the United Kingdom a series of films on the industrial uses of electricity. Ask for a catalogue.

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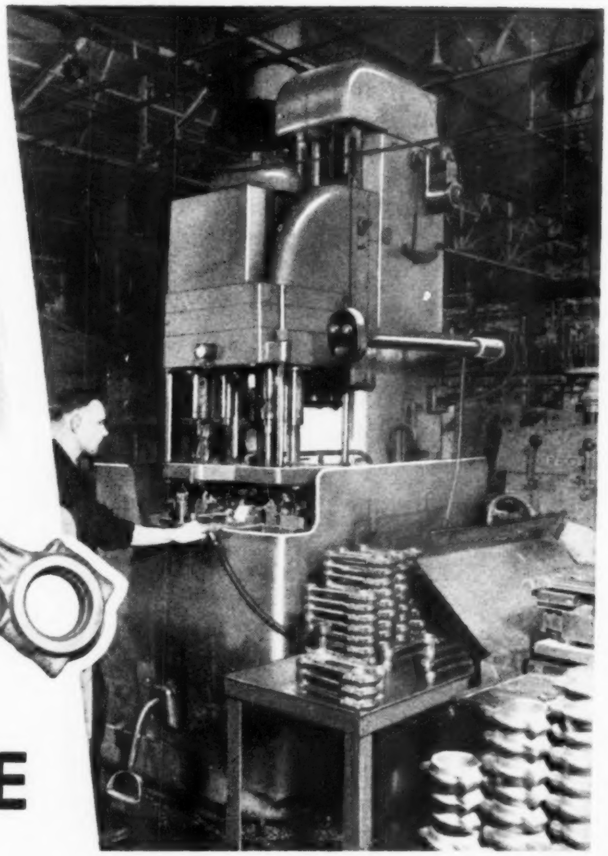

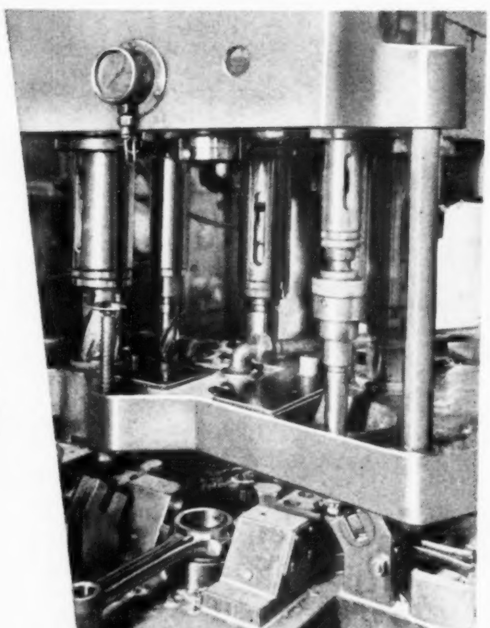

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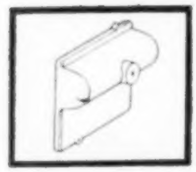

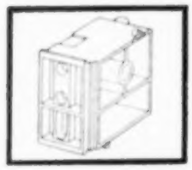

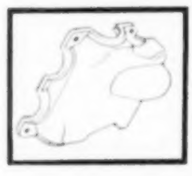

60 amp. Input Junction Box moulded by A.E.I. Plastics (Aldridge) Ltd. for Siemens Edison Swan. The four moulded components are shown in the inset drawings.

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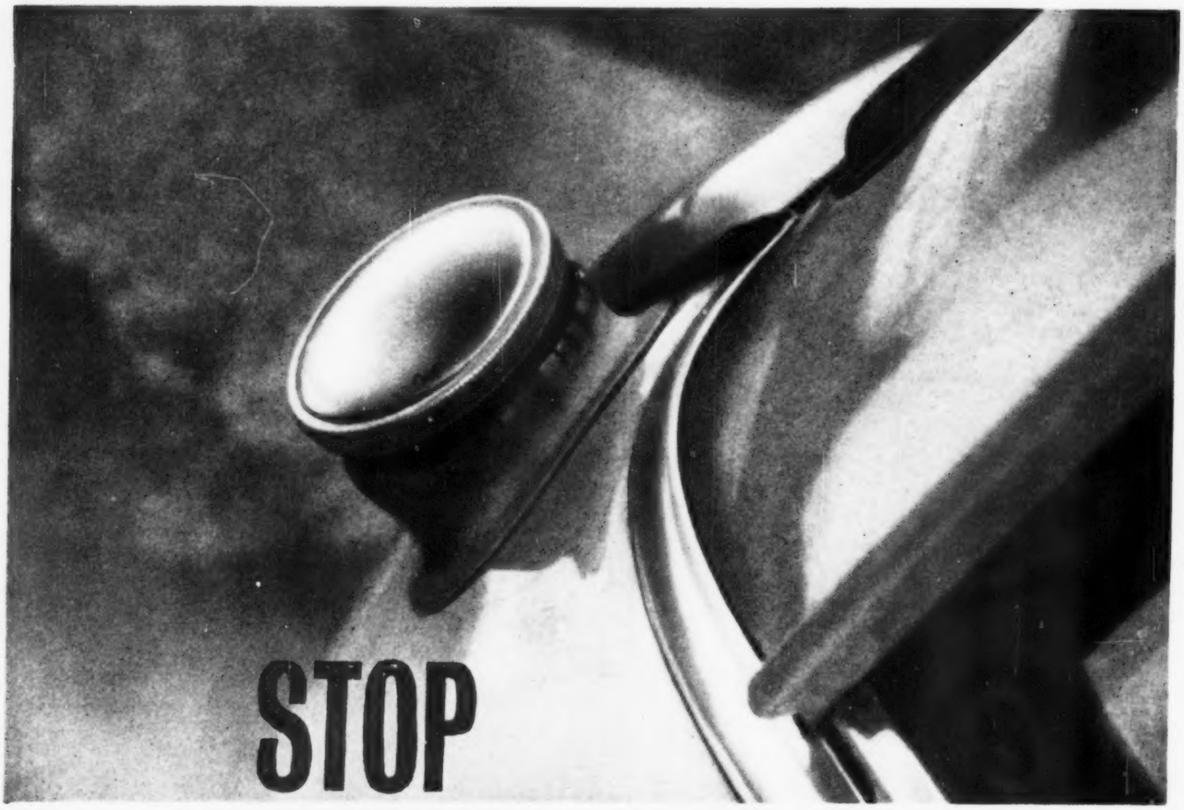

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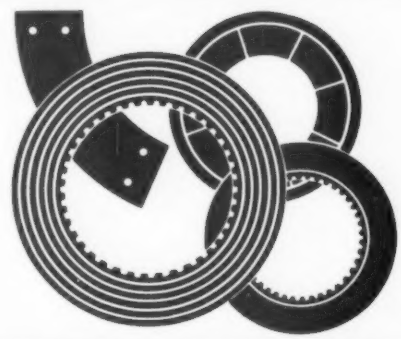

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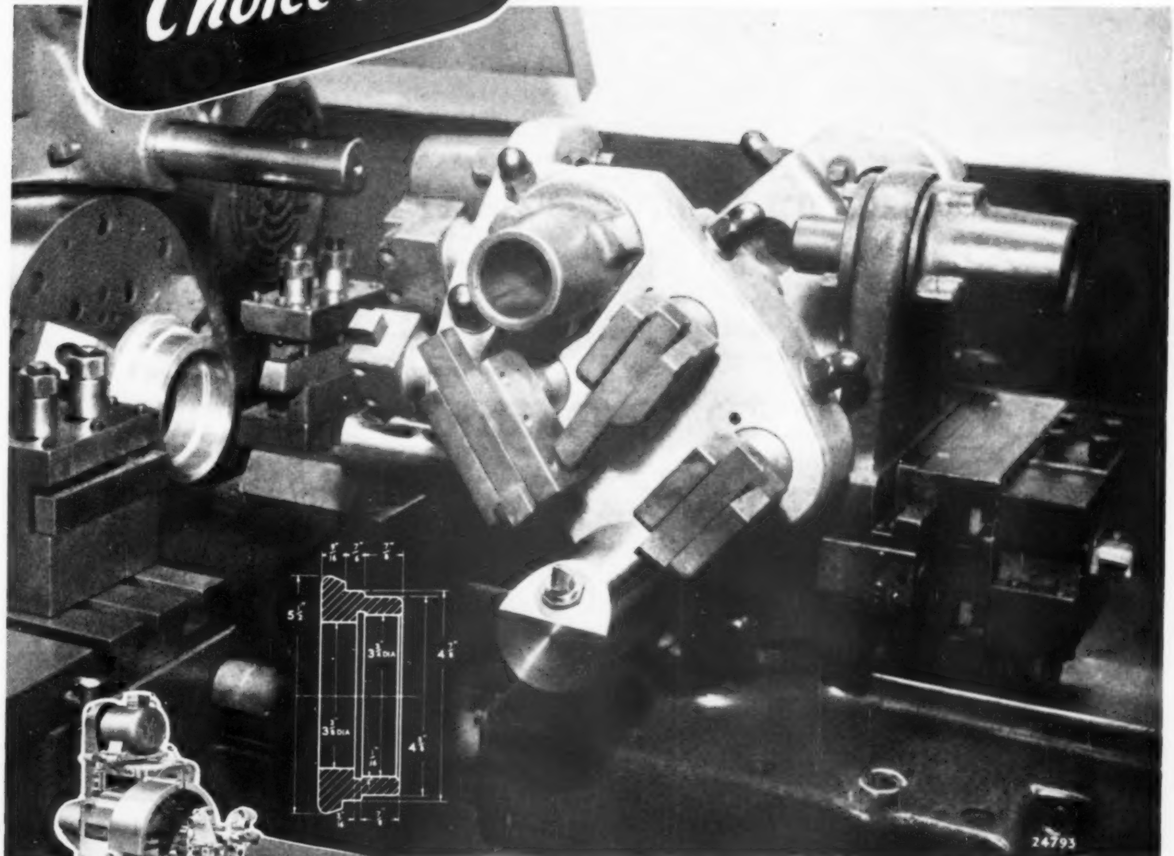

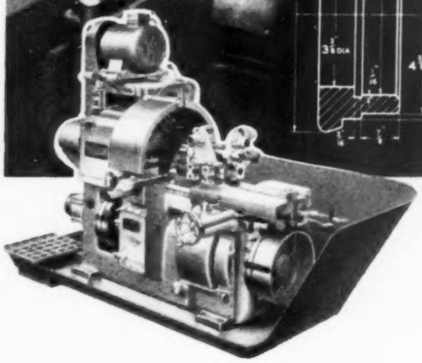

Tooling arrangement on a Herbert No. 3A Auto lathe for the production of thrust plates from 0·45% carbon-steel forgings in a total time of 3·16 minutes each.

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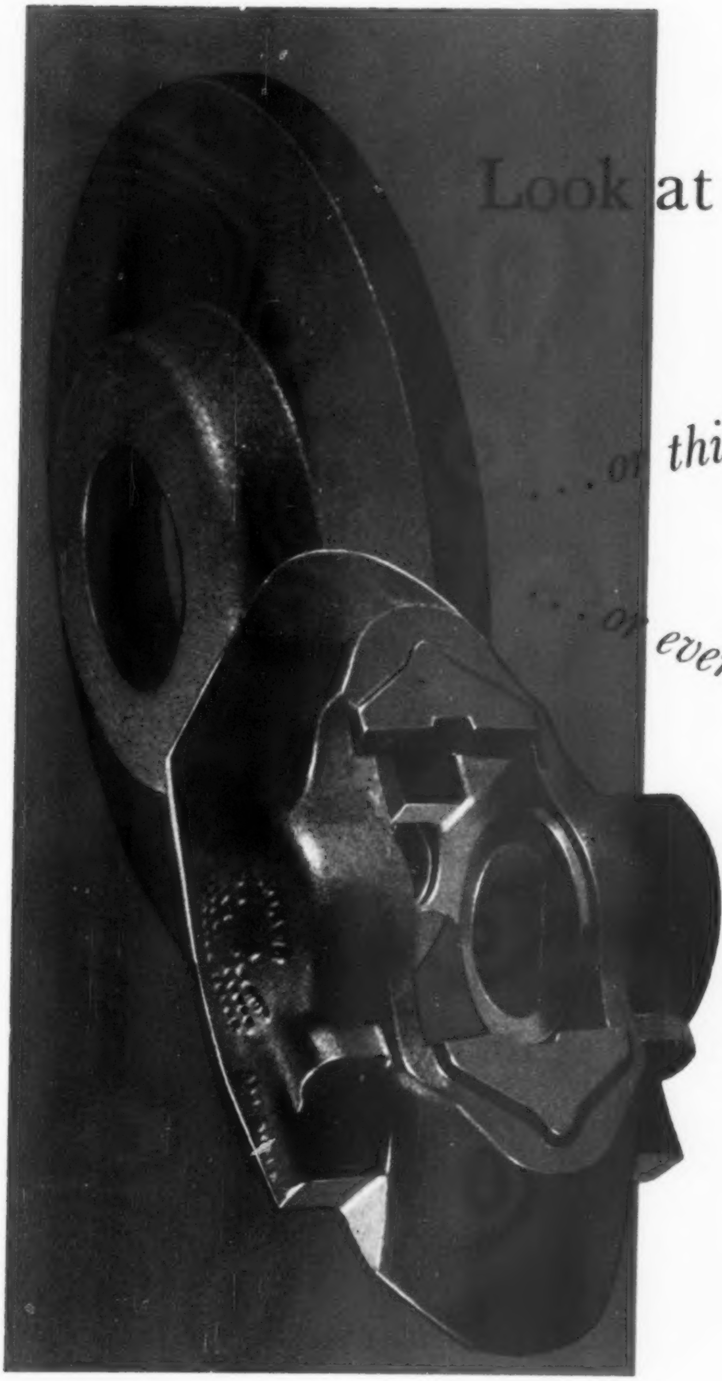

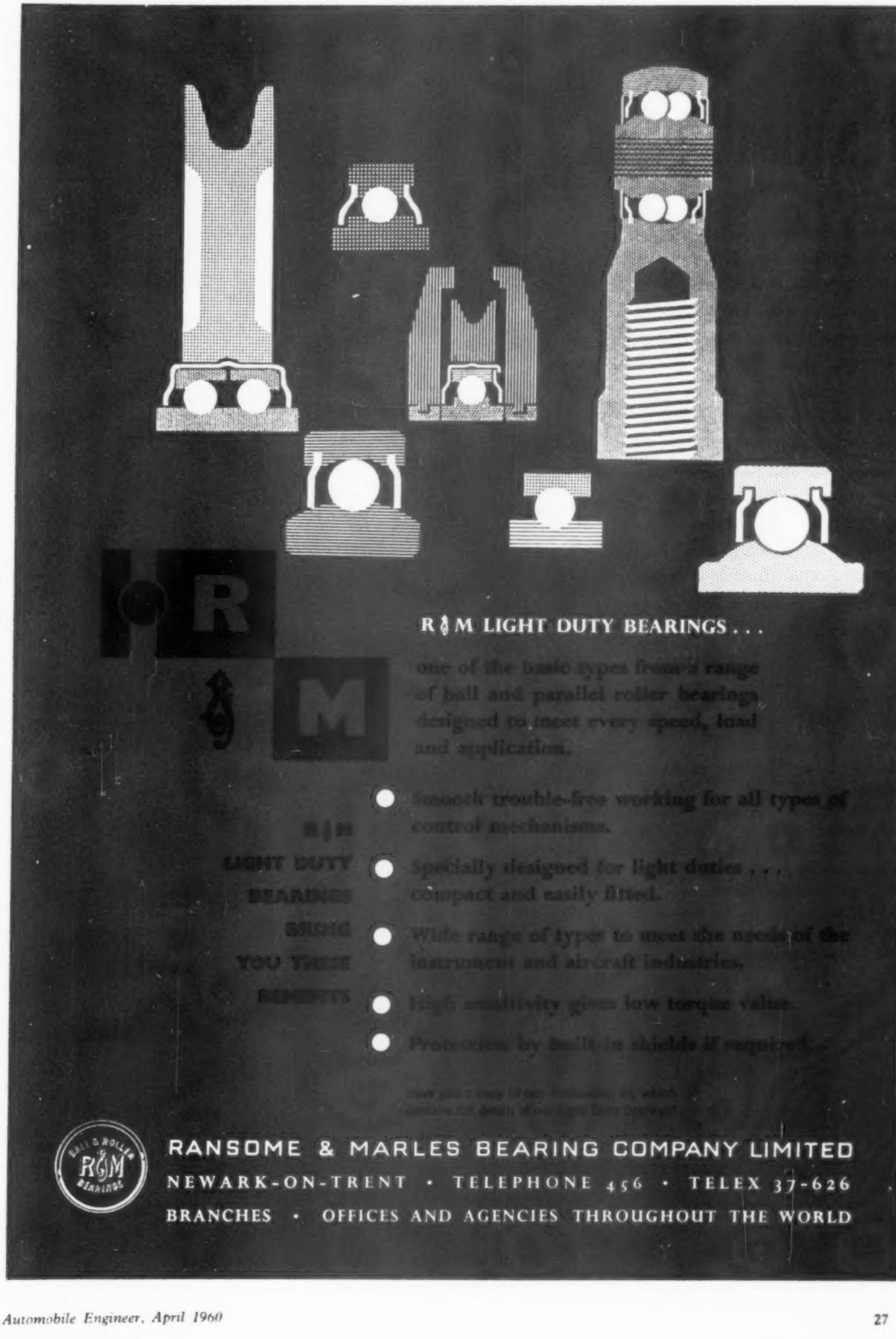
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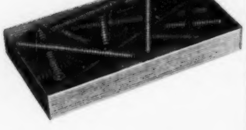

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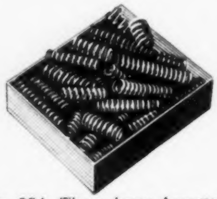

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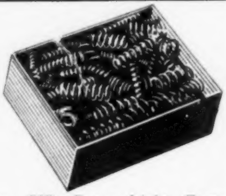

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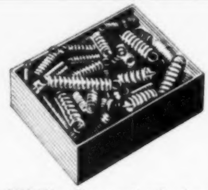

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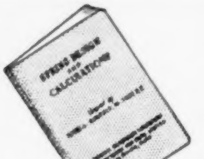

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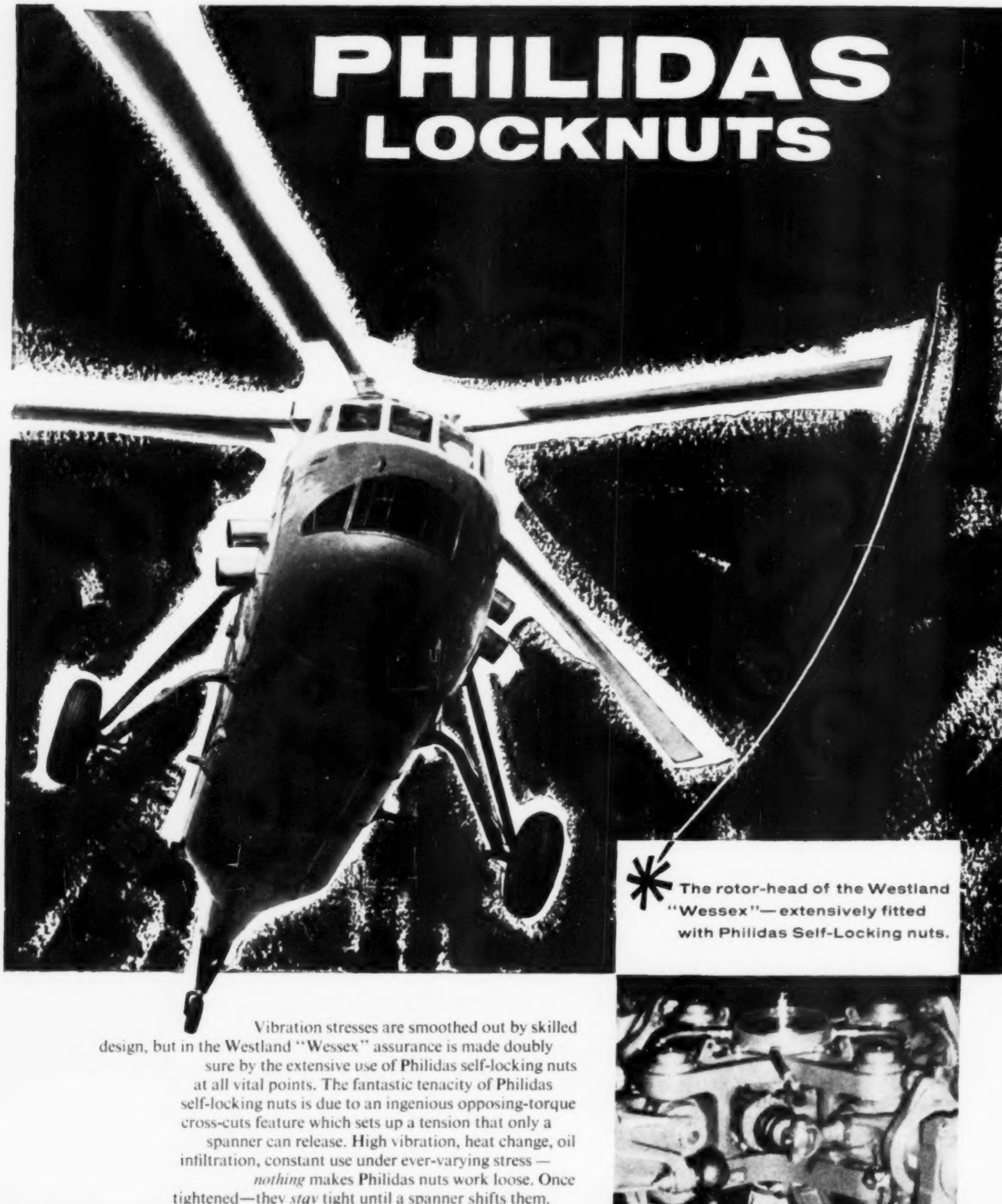

* The rotor-head of the Westland "Wessex"—extensively fitted with Philidas Self-Locking nuts.

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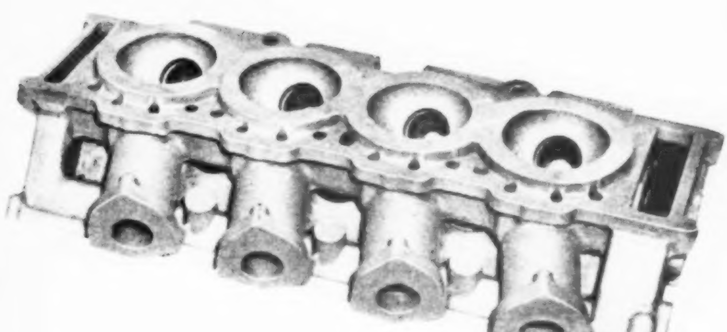

Cylinder Head, sand cast in Hiduminium RR.50 for Rubery, Owen & Co. Ltd.

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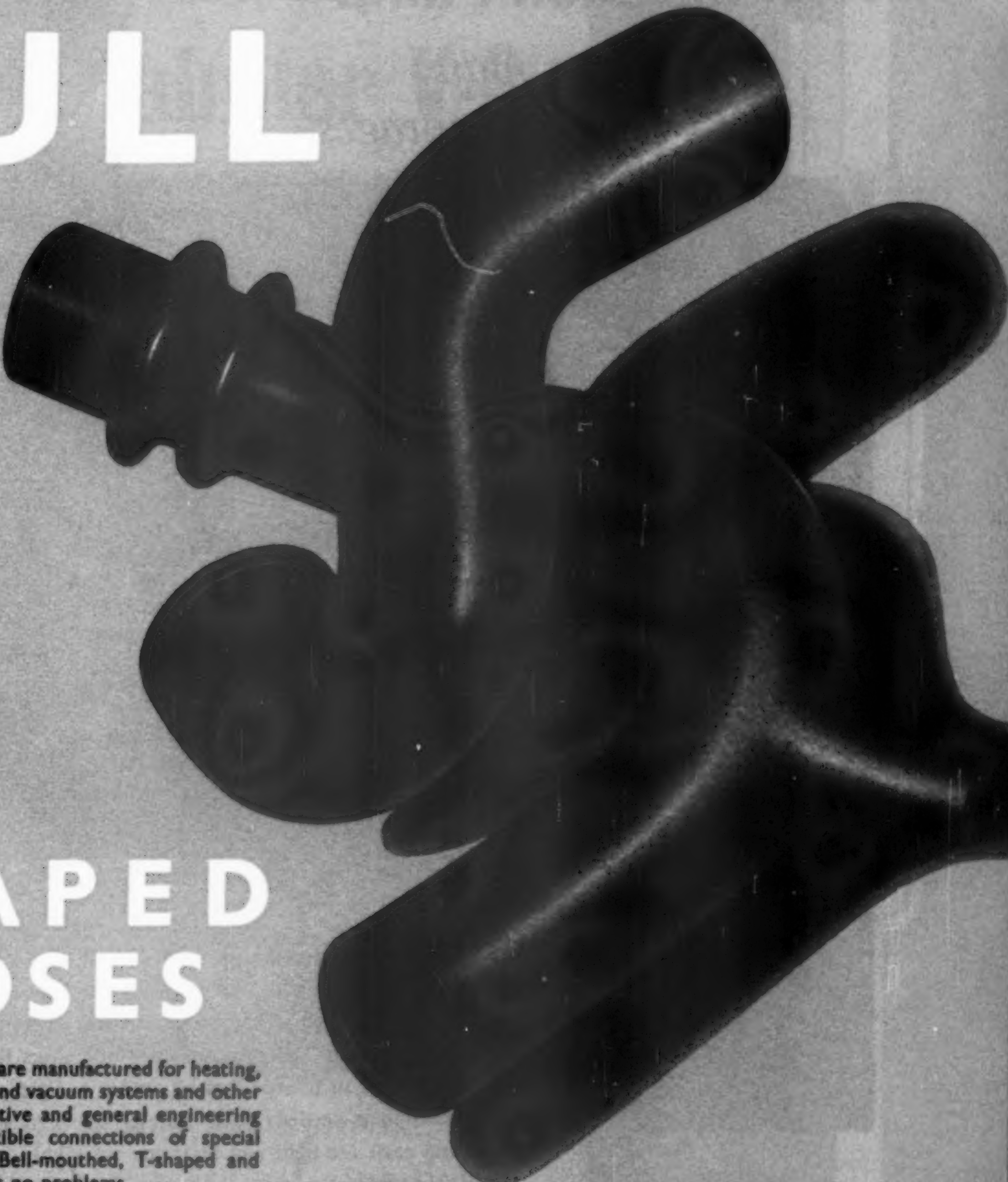

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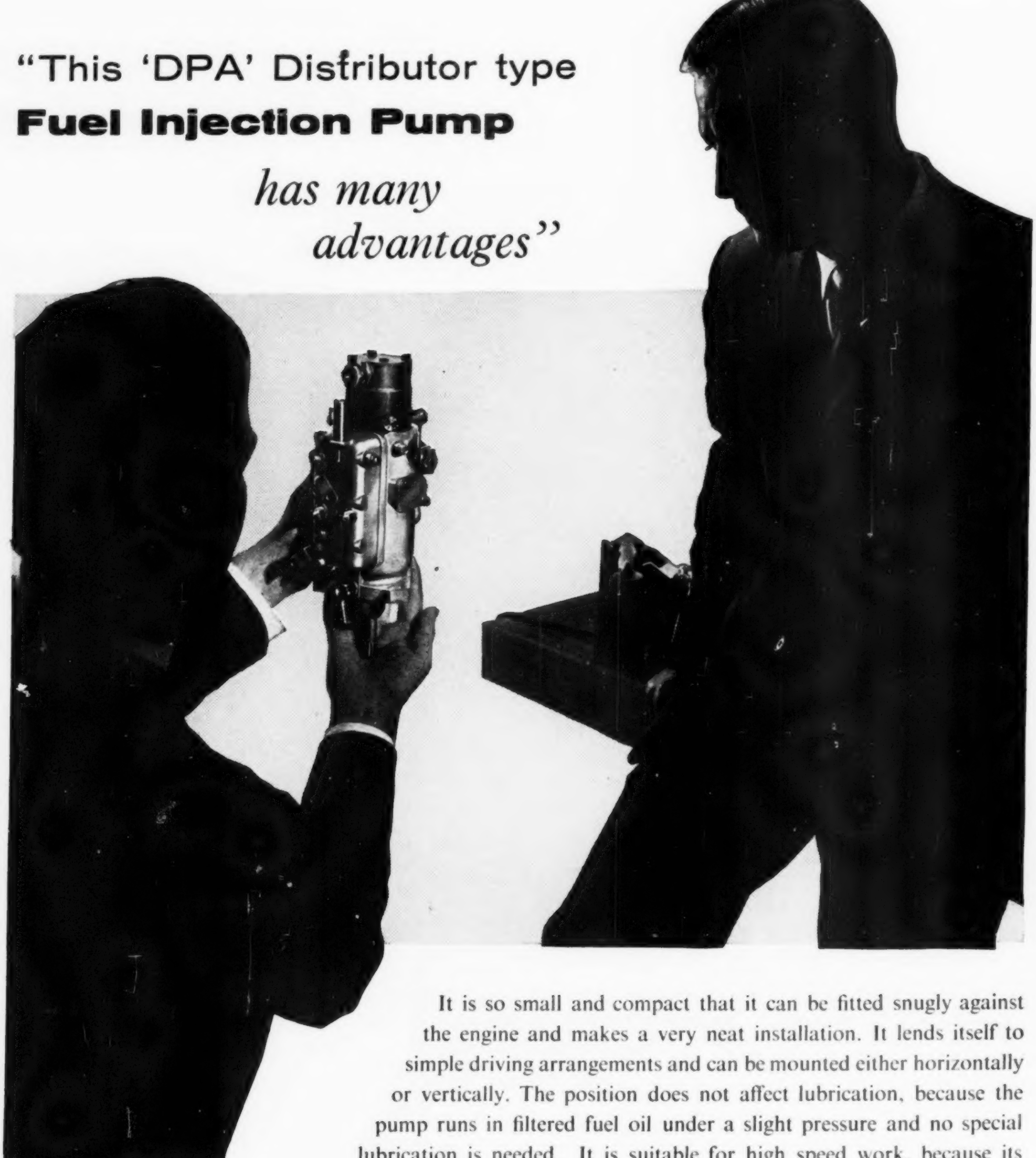
has many advantages"



It is so small and compact that it can be fitted snugly against the engine and makes a very neat installation. It lends itself to simple driving arrangements and can be mounted either horizontally or vertically. The position does not affect lubrication, because the pump runs in filtered fuel oil under a slight pressure and no special lubrication is needed. It is suitable for high speed work, because its rotating parts are light and there are no highly loaded springs. There is much more besides . . .

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AP 975

Illustration by courtesy of "The Motor"

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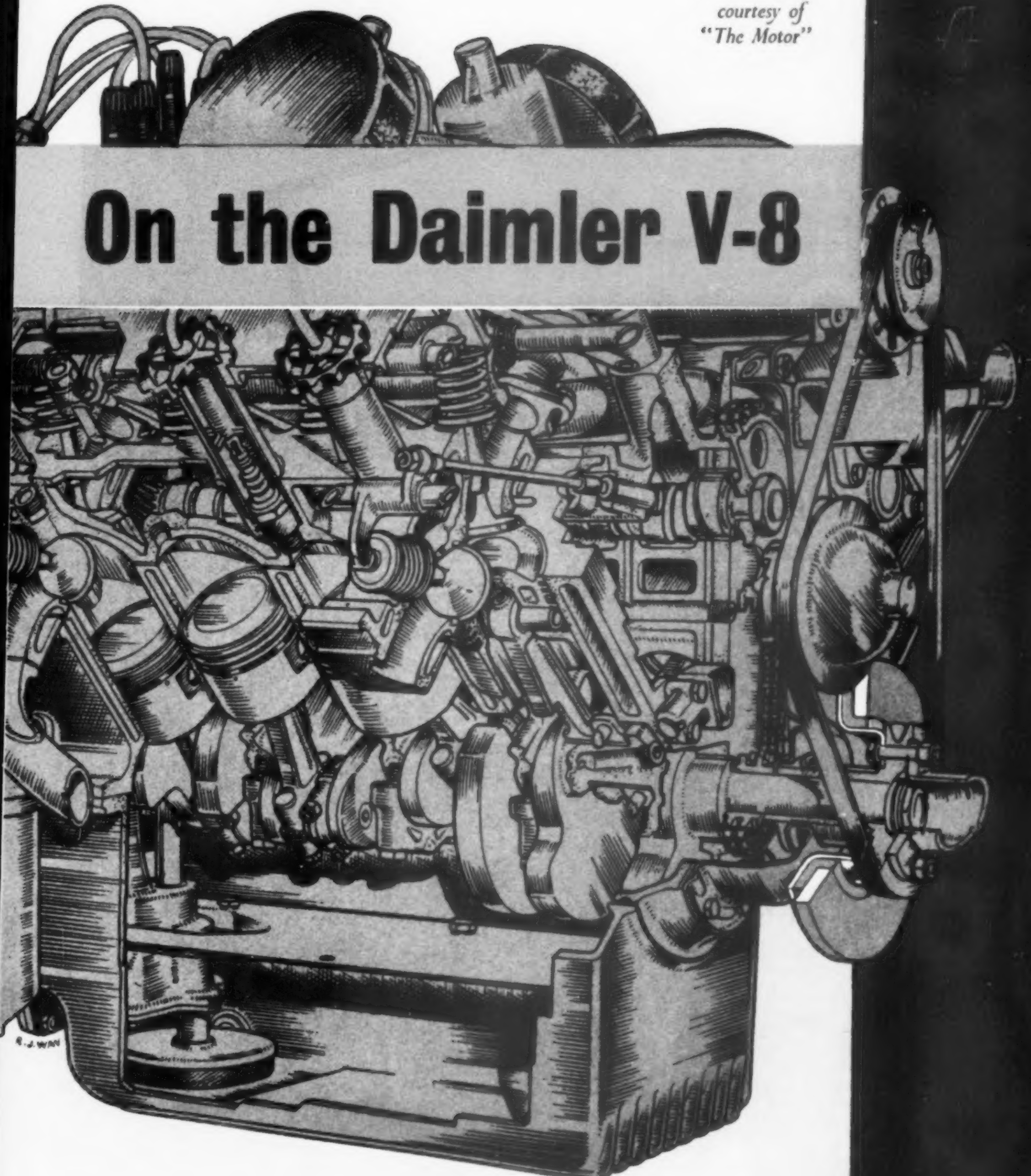

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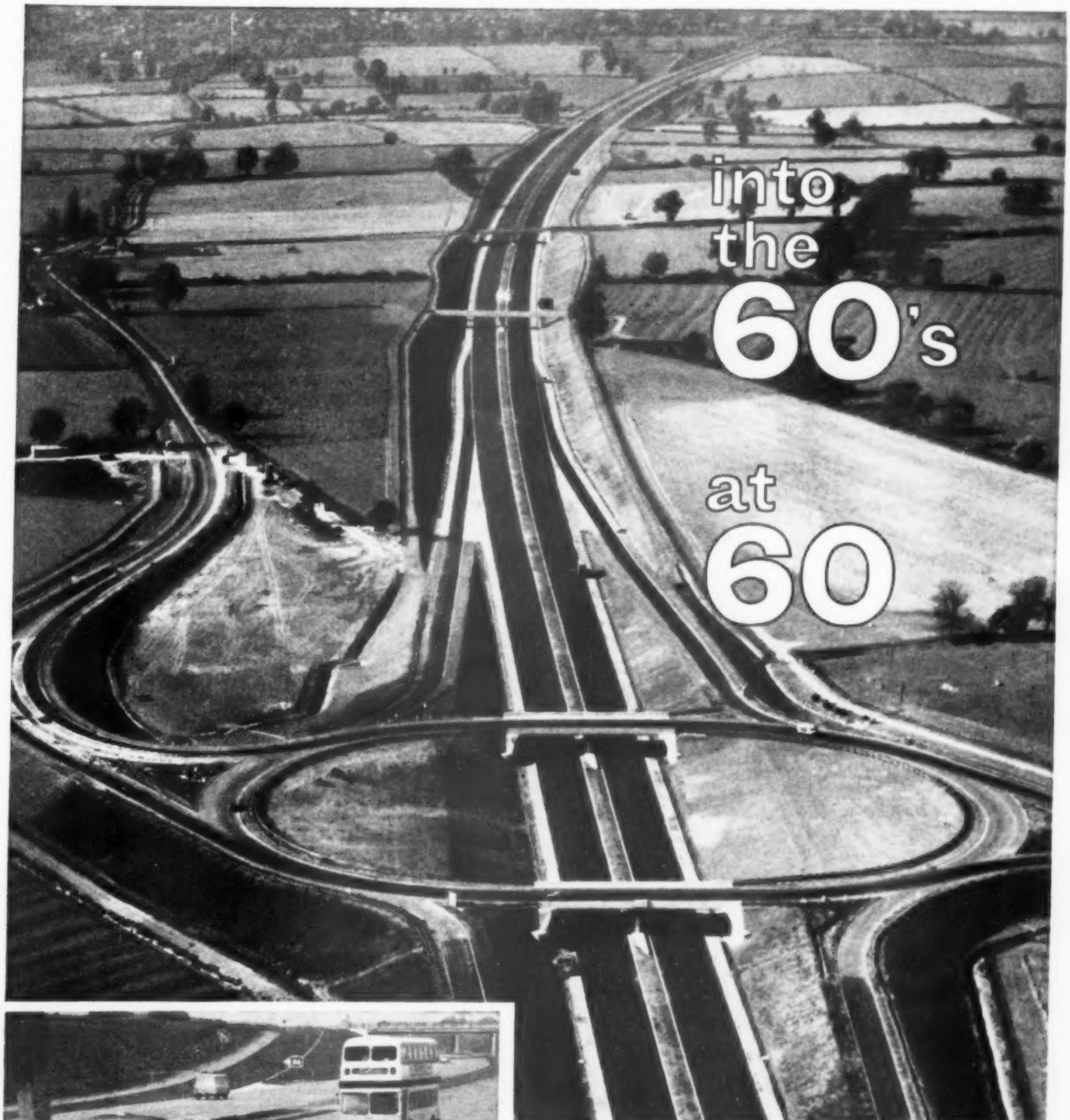
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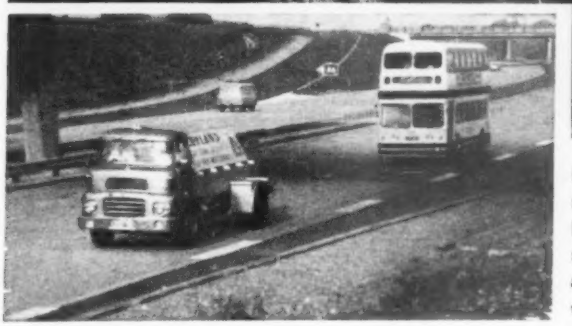

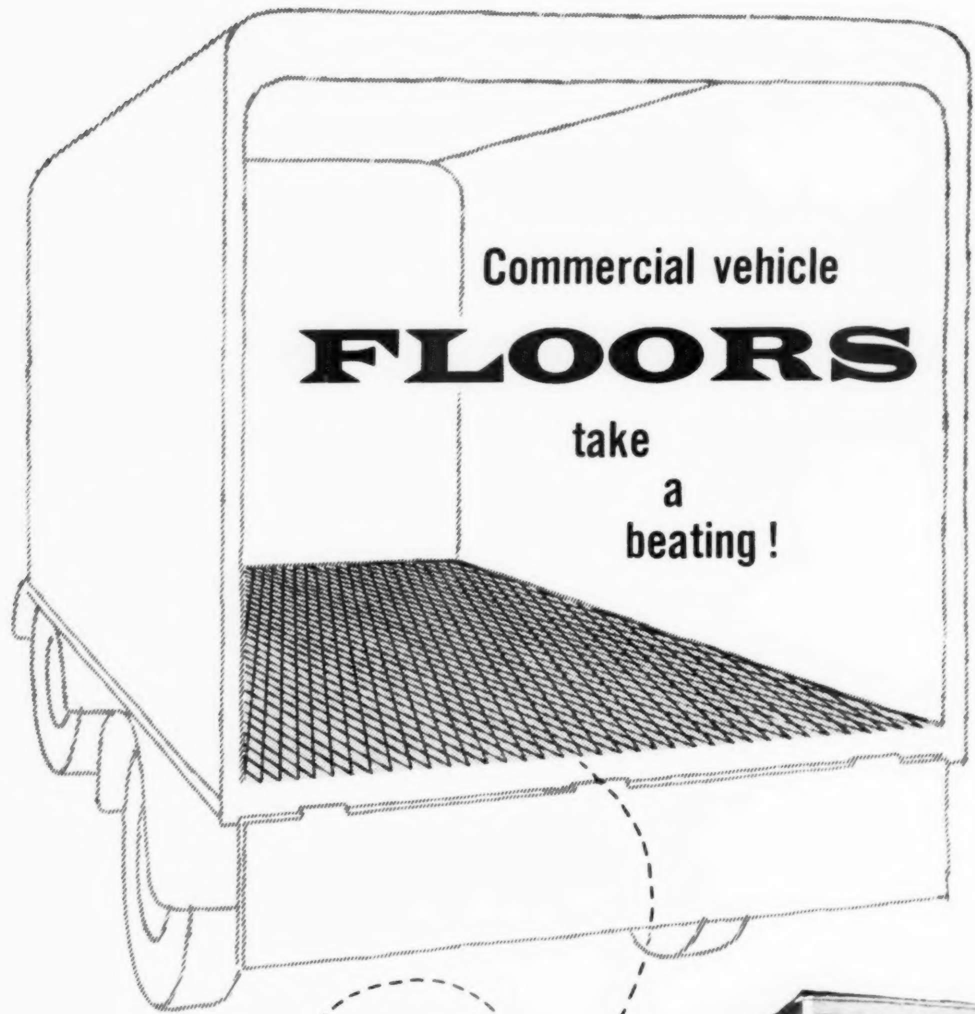

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This newly developed self-conforming grommet, because it is immediately self-locking against the elements, is the solution to many of an engineer's sealing problems.
Any one size will not only accommodate itself to a variety of mounting plate thicknesses, but (designed for cable or control rod) will take these in a variety of sizes and be weather-, water- and dust-proof at a variety of angles to the cable or rod.

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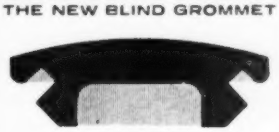
THE NEW BLIND GROMMET



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THE NEW DESIGNED GROMMET

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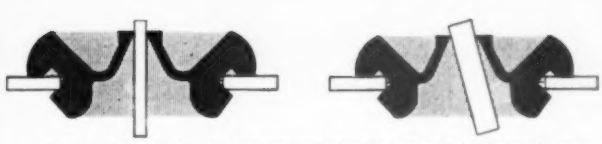
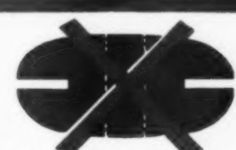
Note how when sprung into position the grommet provides a perfect double seal by its own permanent pressures. The angled groove also creates a tight pressure hold on the metal plate.

In the cable grommet variety the same double pressure seal is created, allied to tight seal on various diameters of cable. This new grommet gives sound sealing at *all* vital points.



A useful feature of this cable grommet is that by reason of the designed taper of the cable entry and the flexibility of the web, a considerable angle of cable entry and a variety of cable size are possible. This avoids necessity for special grommets with angled bores.



In the conventional grommet, only one thickness of plate and only one size of cable can be accommodated. No effective seal is afforded by the parallel groove.

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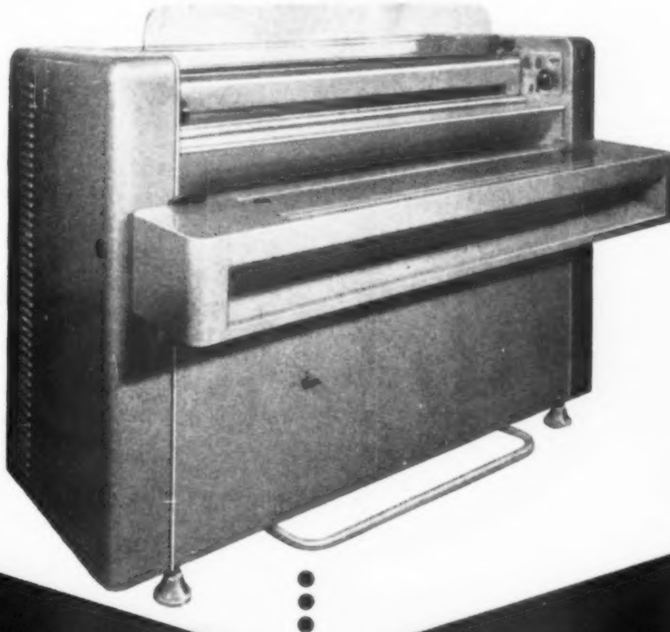

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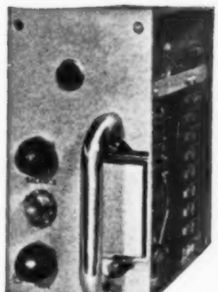

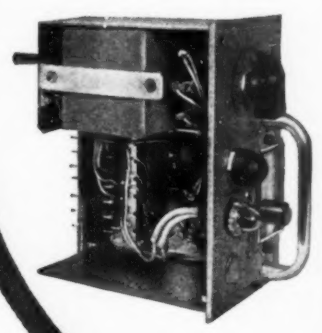

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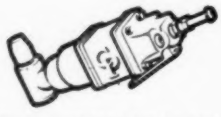
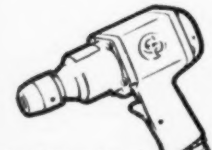
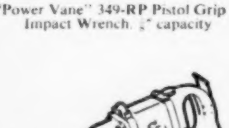


"Power Vane" 337-RAN 90° Angle Impact Wrench. ½" capacity

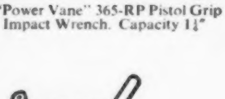


"Power Vane" 344-RS Side Handle Impact Wrench. ¾" capacity

"Power Vane" 349-RP Pistol Grip Impact Wrench. ¾" capacity



"Power Vane" 349-RAN 90° Angle Impact Wrench. ¾" capacity



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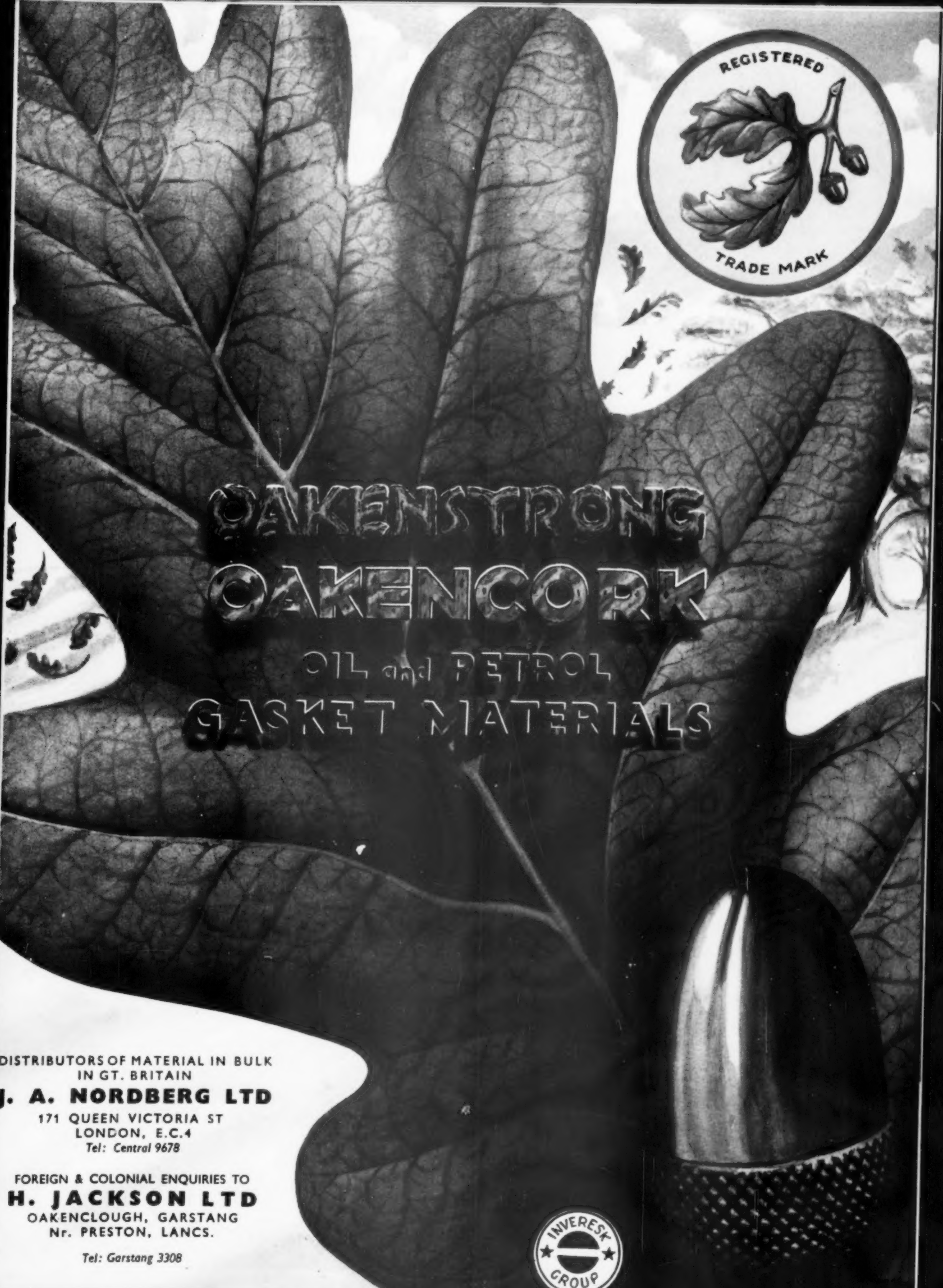
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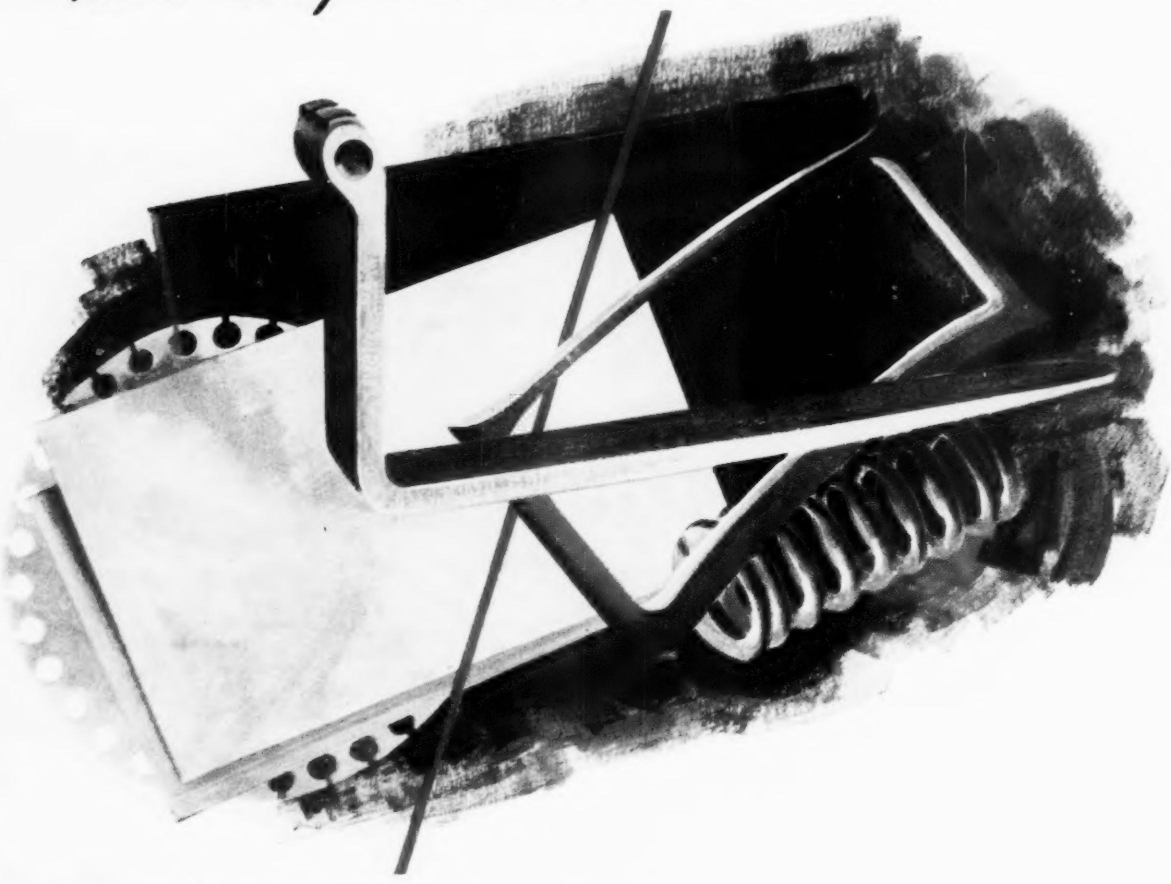

think of R·T·B

The steel frames of their car seats are probably pressed from RTB sheet, so are the bodies; the springs of the independent suspension may be made of our alloy steel; so are the 'strong arms' of the fork-lift trucks used by the motor-car manufacturer.

And at home they are served by tinplate, bright sheets and black steel sheets. Hardly a household has less than a hundred examples; and not far away are RTB laminations, generating electricity or turning it into power.

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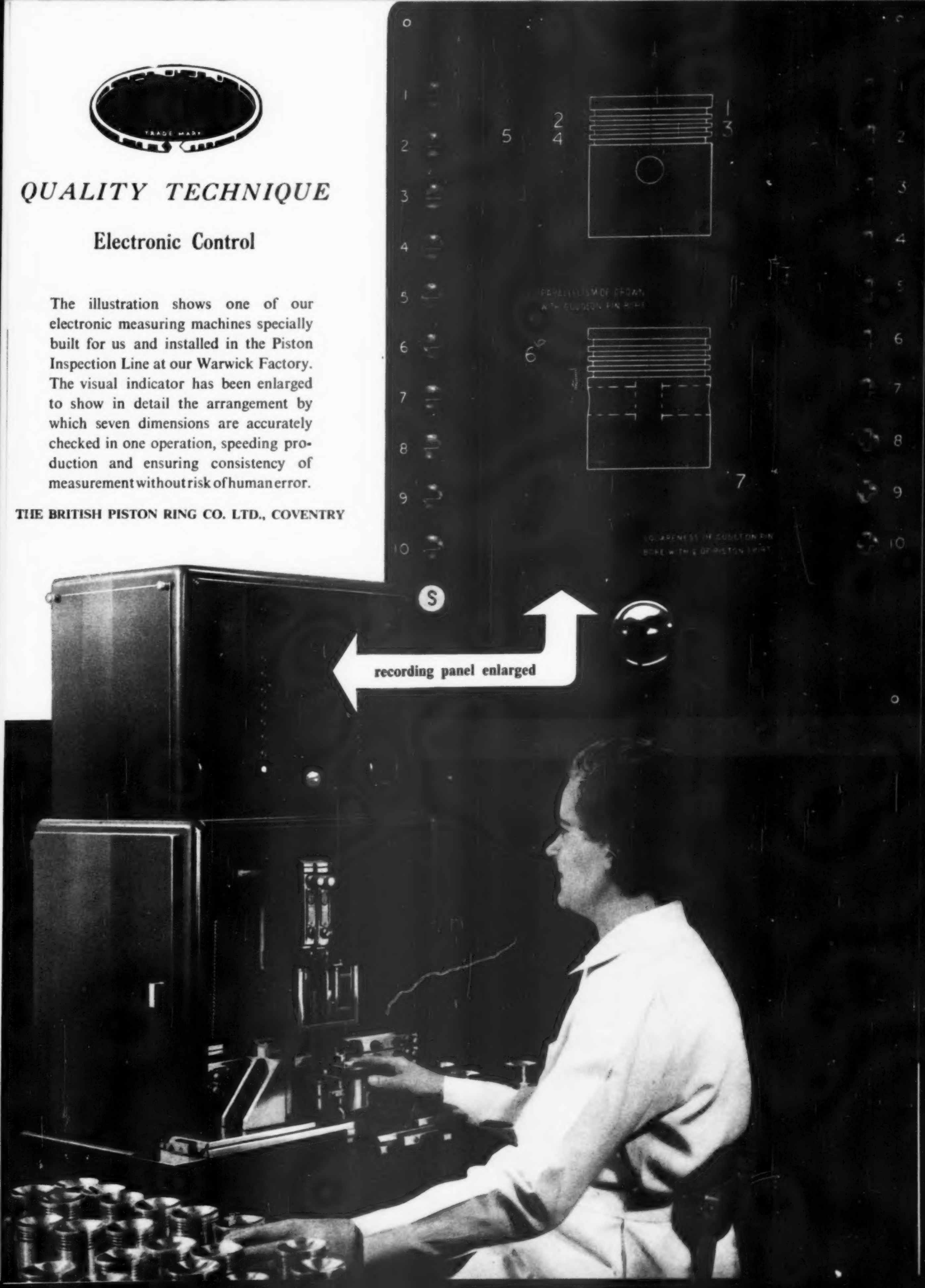

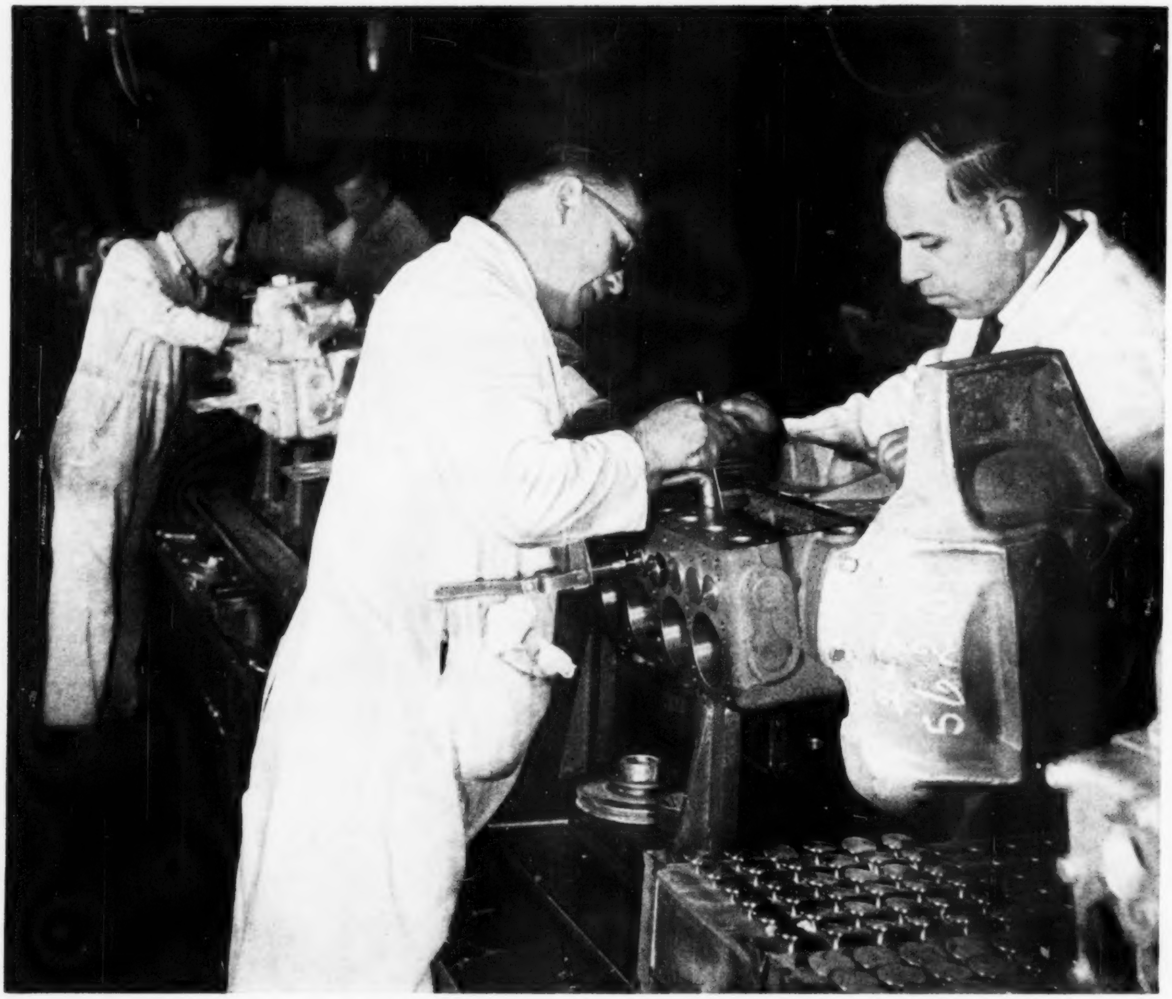
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AUTOMOBILE ENGINEER

CONTENTS

CARY LAMINAIRE SUSPENSION IS DESCRIBED IN THIS ISSUE. THE SYSTEM GIVES A VARIABLE RATE BY THE USE OF A CANTILEVER SUPPORT SPRING, BEARING ON THE MAIN SPRING MASTER LEAF

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VOLUME 50 **NUMBER 4** **APRIL 1960**

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DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

The Key to Successful Development

IN development work, there is scope for considerable savings in cost and time, as well as for improving accuracy and effectiveness. The key is the use of modern methods of instrumentation. But because of the ever increasing complexity of electronic and other devices now coming into use, instrumentation technique is a field that can be dealt with adequately only by specialists. This fact has been given emphasis recently by the introduction of the DIDAS telemetric instrumentation system, described in last month's issue of *Automobile Engineer,* and it began to be apparent as soon as computers were introduced for application to design and development work.

Today by far the most difficult problem to be solved is that of using to the greatest advantage the knowledge of instrumentation engineers. Although they have the capacity for working wonders so far as obtaining information with their equipment is concerned, they cannot provide the data the automobile engineer requires unless they know in considerable detail what information is likely to be of use to him; moreover, unless the automobile engineer himself fully appreciates what can be done by instrumentation, he is, of course, not in a good position to specify what he wants. The problem is further complicated by the fact that development work inevitably involves much groping in the unknown.

It follows that the requirement is for instrumentation specialists with a wide experience in the automobile engineering sphere, and obviously such people cannot be conjured up at short notice. Moreover, since the amount of instrumentation required by individual vehicle and component manufacturers is relatively limited, it will be difficult for their own personnel to gain the necessary experience. Therefore, in Great Britain at least, advantages could be gained by more extensive consultation with the instrumentation department of the Motor Industry Research Association. This would enable the personnel of the Association to broaden even further their knowledge and experience, with consequent benefit to all. There is also scope for the interchange of information between industries : for example, telemetric systems have been in use for some time in missile development, and there are other modern scientific measuring techniques—applied in, among others, the aircraft industry—that could be employed to advantage by the automobile manufacturers.

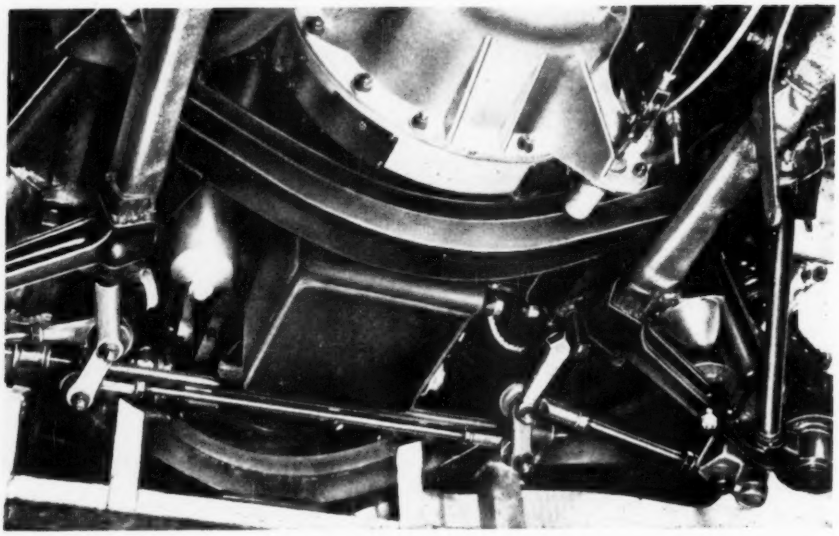
Naturally, there could be opposition to the idea of using M.I.R.A. as a clearing house for the pooling of manufacturers' experience, especially where individual firms feel they have perfected their own methods of development, but the field of instrumentation is undoubtedly too complex to be dealt with adequately by individual effort. Even if the pooling of knowledge were to lead to some small temporary losses in advantages, these will be more than offset by gains that inevitably will also be made in other directions. So far as results and lessons are concerned, there need be no fears—the Motor Industry Research Association is renowned for its discretion.

Improvements are constantly being made in respect of techniques of measuring, recording and analysis of results, and it is becoming increasingly difficult for engineers who are preoccupied with day-to-day mechanical problems to keep abreast of these advances. An example is the piezo-electric vibration strain gauge produced by the Brush Crystal Co. Ltd., of Southampton—this is a completely new application of the piezo-electric principle. The action of these gauges can be reversed: that is, by supplying an alternating current to them, they can be made to vibrate mechanically at the frequency of the applied current.

Another example is strain measuring devices that have high gauge factors. These gauges have been developed by Bell Laboratories, in the United States of America. Factors as high as 150 for germanium and 175 for silicon units have been achieved, as compared with factors of 2 to 4 for the conventional resistance strain gauges. It would also appear that there is scope for the application of gyroscopic instruments for the investigation of roll, pitch and yaw of motor vehicles, and especially of transient phenomena that occur during manoeuvring.

So far as electronic equipment is concerned, there have been misgivings in the past as to its reliability, but the introduction of transistors has changed this situation. Recent progress with more complex systems has led to the introduction of equipment capable of coping with very large numbers of measurements simultaneously. Other developments have made it possible to process these data with equal despatch. If these facilities are to be used to the best advantage, extensive planning of development work is necessary, and it is obvious that those who do the planning will be required also to perform the function of intermediaries between the instrumentation specialists and the experts in the automobile mechanical engineering field. That this subject merits careful study is obvious, since the benefit to be gained from an entirely new and systematic approach is a better product at lower cost and enhanced potential for meeting competition throughout the world.

ROVER 3 LITRE



This view from beneath reveals several interesting details of the Rover front suspension. The laminated torsion bars fit into holes in the lower transverse links, which are braced by radius arms. In the steering linkage, the joints of the three-piece track rod are sealed

Part III: Analysis of the Rear and Front Suspension Systems, also the Steering Layout, the Brakes and Electrical Equipment

ALTHOUGH otherwise orthodox, the rear suspension of the Rover 3 Litre car is noteworthy for having no metal-to-metal contact with the body structure. It is well known that cars of unitary construction are more susceptible to the transmission of road-generated noise to the interior than are those with a separate chassis frame. To attain the degree of quietness traditionally associated with Rover cars, advanced methods of insulation were therefore necessary, and those adopted have proved markedly effective.

The semi-elliptic springs have a length of 49¼ in between attachment points. To provide a slight roll steer effect, their front ends are mounted approximately 3¼ in lower than the rear ends. The axle is situated at the middle of the springs, the action of which is controlled by hydraulic dampers of the telescopic type.

Each spring has seven main leaves of En.45A steel and, following normal practice, the master leaf is rather thicker than the others. The actual thicknesses are: master leaf, 0·218 in; other leaves, 0·171 in. In the normal static, laden position, with three persons in the car, the springs have a slight negative camber, of the order of ⅜ in. Below the main leaves is a helper leaf, with a thickness of 11/32 in; it is of the same material as the other leaves and has virtually no camber. For the last 3 in of each end, it is tapered in thickness, on the upper surface, to give a progressively rising rate as the load increases. The combined spring rate rises from 100 to 220 lb/in, as the helper leaf comes into action. A periodicity of 85 c/min is quoted by The Rover Co., and the roll centre height is 10·1 in.

All the leaves are clamped together in the middle by a bolt, and there are additional clamps round all the main leaves, beyond the ends of the auxiliary leaf, and also round the upper three leaves near the extremities of the spring. Welded to the underside of the axle tube, near each end, is the seating bracket for the spring. It is fabricated from 3/16 in steel plate and is of channel shape in front elevation, with a medial reinforcing flange parallel to the other two. The flanges are extended ahead of the axle to form the lower anchorage of the damper.

U-bolts of conventional design secure the axle to the springs, which are 42 in apart. These bolts seat directly on the tubular portions of the axle casing, and their ends pass through retainer plates. Bending of the intermediate portions of the plates under bolt tension is resisted by flanges at the front and rear ends. A hole in the middle of each plate provides clearance for the nut of the medial clamping bolt of the spring.

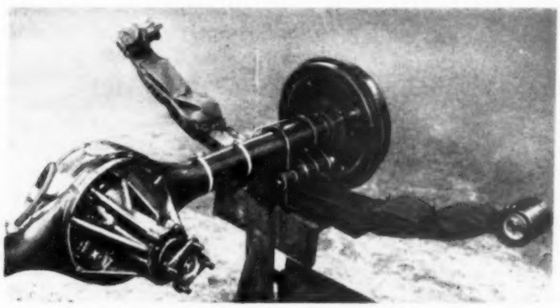
Although the Metalastik attachments of the spring to the support brackets on the body were described in the April 1959 issue of *Automobile Engineer*, a brief recapitulation here is merited. At the front end of the spring, the eye of the master leaf houses a Metaxentric bush. As the name Metaxentric implies, the axis of the inner sleeve of the bush, in the unloaded state, does not coincide with that of the outer sleeve but is offset above it. The heaviest components of the loads are thus taken in compression on the greatest thickness of rubber, and the design is such that vertical flexibility is considerably greater than longitudinal flexibility. Interesting details of this bush are the unusually thin outer sleeve and the slotting of the rubber at its thinnest section, to avoid stressing it in tension.

At the rear of the springs, Contrasonic shackles are employed in place of metal shackles. In the Contrasonic unit, rubber is used in combined compression and shear on bump, and in tension and shear on rebound movement. In addition to its sound insulating properties, this ingenious design provides greater lateral stiffness than do most rubber bushed shackles of normal design. A further advantage is that the unit is bolted to the master leaf of the spring, which, therefore, does not have to be formed into an eye.

The Contrasonic shackle comprises a steel bracket of triangular section to which are bonded two bobbins of rubber, moulded as a unit: the base of the triangle sits on the spring, and the bobbins project perpendicularly from the other two sides. To the outer end of each bobbin is bonded a plain washer carrying a captive bolt for attachment to the vehicle. A bridge of rubber between the bobbins forms a bump stop, to limit the upward deflection of the shackle.

A rubber bump stop for the axle is bolted to each side member of the body rear structure. The double-acting dampers, of Woodhead Monroe manufacture, each have a bore of 1 in diameter, and are sited immediately behind the rear seat squab. They are installed with a slight forward inclination, so that they are substantially at right angles to the spring, but without any lateral inclination. Their upper ends are attached, by rubber sandwich type end fittings, to channel section brackets welded to the body side members and the inside panels of the wheel arches, which close them to form box sections. At its lower end each damper has an overhung attachment, of rubber bush type, to one of the spring seating brackets on the axle.

The bushes are of split design, and each half-bush is of tapered section to provide location for the eye of the damper. Each flange extension of the spring seating bracket is drilled, and a steel tube passes through the three holes. The tube is welded in position. Its inboard end projects beyond the



Above: The rear springs are gaitered and are mounted on Metalastik Contrasonic shackles and Metaxentric bushes. Right: Extensions of the spring seating brackets form the lower anchorages of the dampers. The tapered upper surface of the helper leaf is clearly visible

bracket to form the inner sleeve of the rubber bearing. Because of the amount of offset of the damper axis from the spring bracket, a distance sleeve is interposed between the bush and the bracket: this sleeve fits over the tube and is welded to the bracket. The bearing is retained on the tube by a large plain washer and a bolt.

Front suspension

Although the front suspension is of largely conventional design, it is entirely new and embodies several interesting features. The outstanding one is probably the use of relatively short laminated torsion bars as the springing medium. Others include the use of ball-joints for the steering swivels, and the employment of lateral arms and diagonal drag links, instead of wishbones, for the lower linkage of the suspension. In terms of reduced running costs and time off the road, the obviation of the need of periodic lubrication is a praiseworthy innovation. Satisfactory cornering stability is obtained by the use of an anti-roll bar, but it is probable that the adoption of a different suspension geometry from that previously employed has assisted in this respect

Each of the front wheels is mounted on taper roller bearings on a stub axle integral with the forged 5030 steel

SPECIFICATION

SUSPENSION: Front, double transverse link, with laminated torsion bar springs, Woodhead Monroe telescopic dampers and anti-roll bar. Rear, semi-elliptic springs, with Woodhead Monroe telescopic dampers.

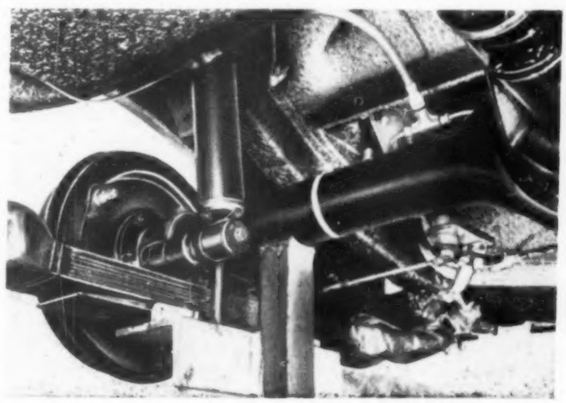
STEERING: Burman recirculating-ball, with three-piece track rod and slave lever. Ratio 17·6 : 1 straight-ahead; 28·5 : 1 on full lock. $4\frac{1}{2}$ turns of steering wheel from lock to lock. Turning circle, 40 ft.

BRAKES: Girling hydraulic, with Girling vacuum servo assistance. Front, disc type, with 10·794 in diameter discs. Rear, drum type, 11 in diameter; shoe width, $2\frac{1}{4}$ in. Total swept area, 413 in². Hand brake actuates rear shoes through mechanical linkage.

WHEELS: Steel disc, with five-stud attachment, 5 in wide rims. Tubeless tyres, 6·70-15 in (standard) or 7·10-15 in (oversize). Pressures, standard tyres: front, 24 lb/in²; rear, 22 lb/in².

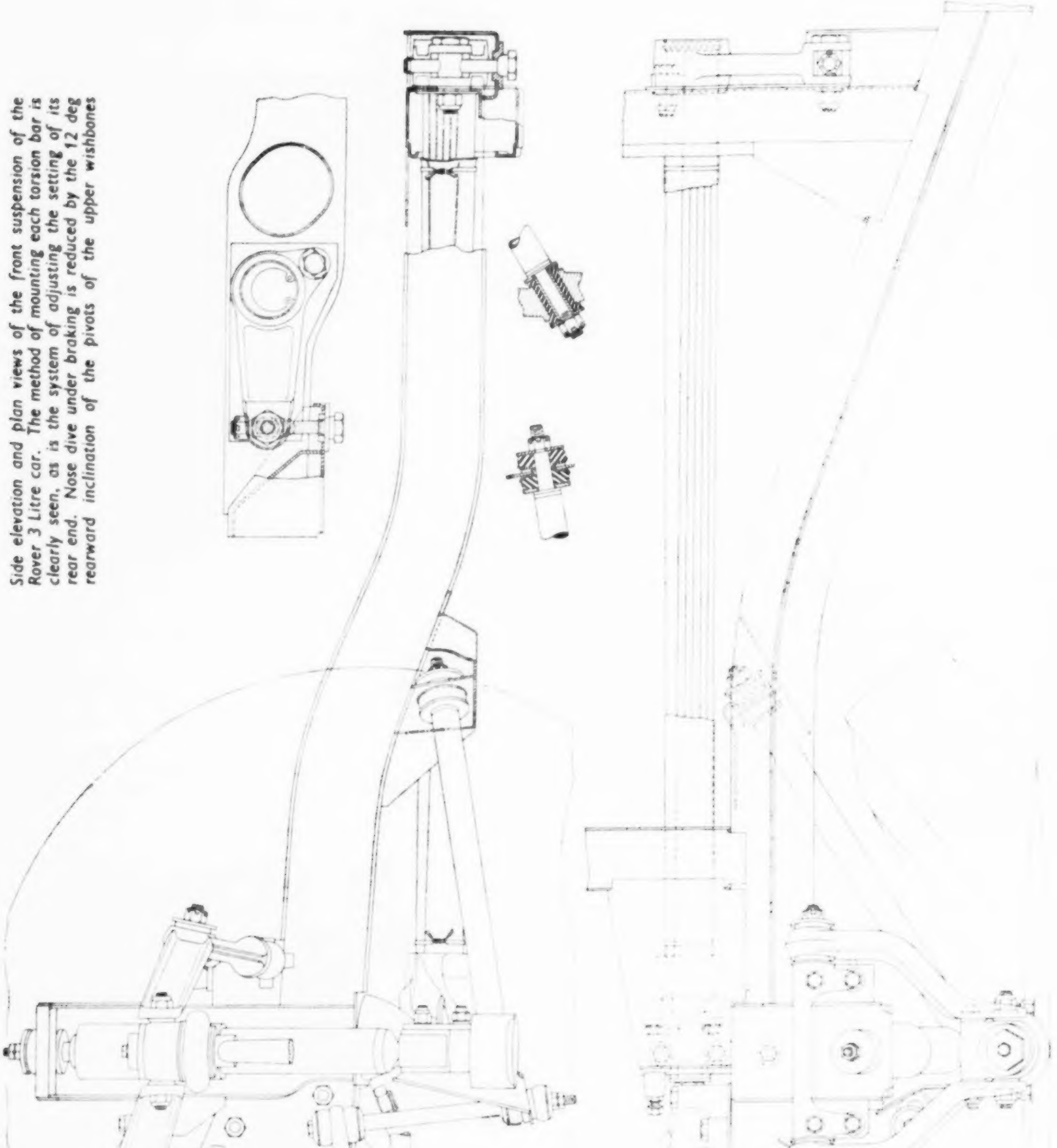
DIMENSIONS: Wheelbase, 9 ft $2\frac{1}{2}$ in. Track: front, 4 ft $7\frac{5}{16}$ in; rear, 4 ft 8 in. Overall length, 15 ft $6\frac{1}{2}$ in. Overall width, 5 ft 10 in. Overall height, unladen, 5 ft $0\frac{1}{4}$ in. Ground clearance, $7\frac{7}{8}$ in. Frontal area, approximately 20·8 ft². Kerb weight of standard transmission model, with full fuel tank, 3,598 lb. Weight distribution: front, 54 per cent; rear, 46 per cent.

swivel member. The upper and lower ends of this member are offset to accept the swivel pins, which have a taper mounting in the member and form part of the ball-joints. These ball-joints cater for both steering and suspension articulation, and are widely spaced, at $11\frac{1}{4}$ in apart, to minimize the stresses from lateral loads. The upper one is a plain spherical bearing, incorporating controlled friction loading through the medium of a compression spring in the housing. A part-spherical head is formed at the top of the swivel pin; this head is of the same radius as the main part of the ball, which is of hardened steel and floats on the pin. The ball seats in a hardened cup that fits into the housing and is clamped in position by the screw-in cap carrying the spring. To exclude dirt and water, and to retain the lubricant, a synthetic rubber gaiter is fitted between the housing and a pressed steel shroud; this is pressed on to



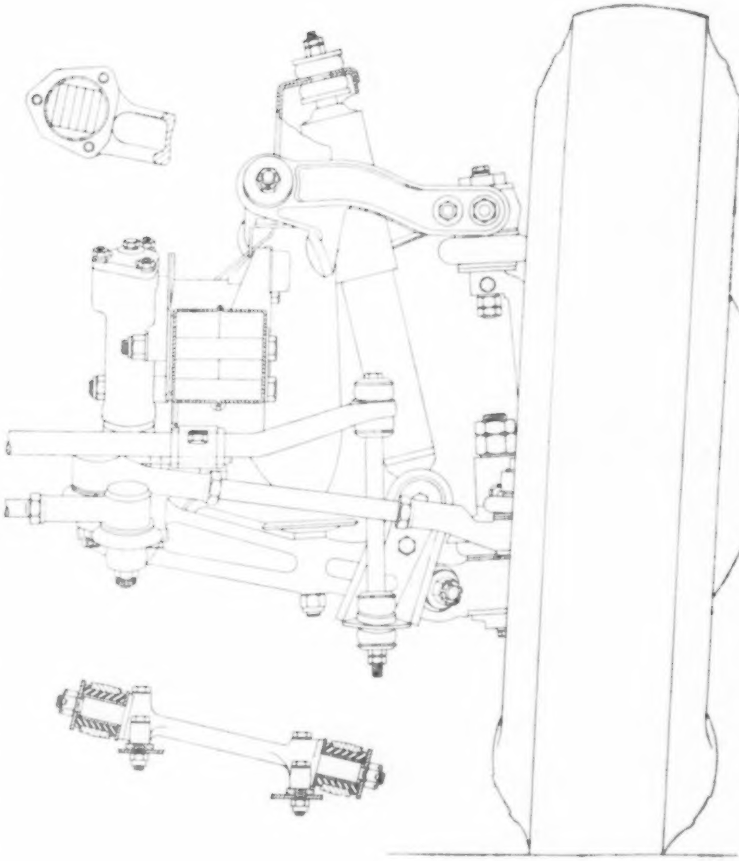
the swivel pin below the ball, the bore of which is counterbored to receive it.

Since it has to support a portion of the vehicle's weight, the lower ball joint is of more massive construction. The ball seats directly in the housing but is not a complete sphere: its lower portion is machined away to form the upper track of a ball thrust bearing, the lower track of which is machined on an integral collar on the swivel pin. A spring abutting on the underside of this collar maintains contact between the ball and its seating during negative loading. It is clear that, with this arrangement, if the ball



Side elevation and plan views of the front suspension of the Rover 3 Litre car. The method of mounting each torsion bar is clearly seen, as is the system of adjusting the setting of its rear end. Nose dive under braking is reduced by the 12 deg rearward inclination of the pivots of the upper wishbones

Each upper wishbone comprises two steel stampings, whereas the lower links are forgings. The steering swivel members are connected to the wishbones and links by ball-joints. Rubber bushes, mounted beneath the side members of the front sub-frame unit, carry the anti-roll bar



were to rotate on the swivel pin axis within the housing, the thrust bearing would not be fully effective in minimizing friction. Any such rotation is prevented by a dowel in the housing wall, which engages a groove on the ball. The sealing method is similar to that of the upper joint but, because the shroud is inverted, the swivel member carries a second shroud that acts as an umbrella over the gaiter and the top of the first shroud.

Wishbone type upper suspension links are employed, and each comprises two 1035 steel stampings of channel section, with the flanges facing outwards. The outer ends of the channels are secured to the housing of the upper ball joint, and the inboard ends of the links pivot on bonded rubber bearings, mounted on a forged member that is bolted to a bracket on the front sub-frame; the diameter of the pivot spindles is $\frac{5}{8}$ in. Plain washers are interposed between the inner sleeves of the bearings and the castellated nuts that clamp them in position. The pivot axes of the wishbones are longitudinal in plan view but are inclined downward towards the rear by 12 deg, in order to reduce the nose dive resulting from weight transfer during braking. A detail of interest is the arrangement of the rebound stops: a short arm extends downward from the inboard end of each link of the wishbone, and terminates in a convex pad, which comes into contact with a rubber buffer on the sub-frame.

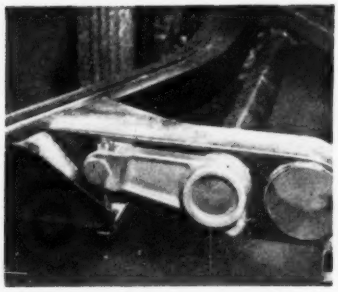
The lower links of the front suspension are I-section forgings in En.19C steel, and their outer ends are enlarged to form the housings of the ball-joints already described. These links are disposed transversely, with a horizontal pivot axis, and each has a large boss at its inboard end. Secured, by three equally spaced bolts, to the leading face of this boss is a flange formed on the $1\frac{3}{8}$ in diameter pivot spindle. The inner sleeve of the rubber pivot bearing is clamped against the flange by a plain washer and castellated nut. Each end of the outer sleeve of the bearing is flanged for location in the housing, which is bolted to the underside of the sub-frame.

Triangulation of the lower links of the suspension, to resist longitudinal loading and brake torque reaction, is provided by obliquely mounted, tubular radius arms, of 1 in diameter. They are fitted between the outer ends of the links, immediately inboard of the ball-joint housings, and brackets below the sub-frame side members. The position of the rear anchorages has been carefully chosen to avoid excessive interference between the natural arcs of movement of the links and the arms, which have an outward inclination of 32 deg.

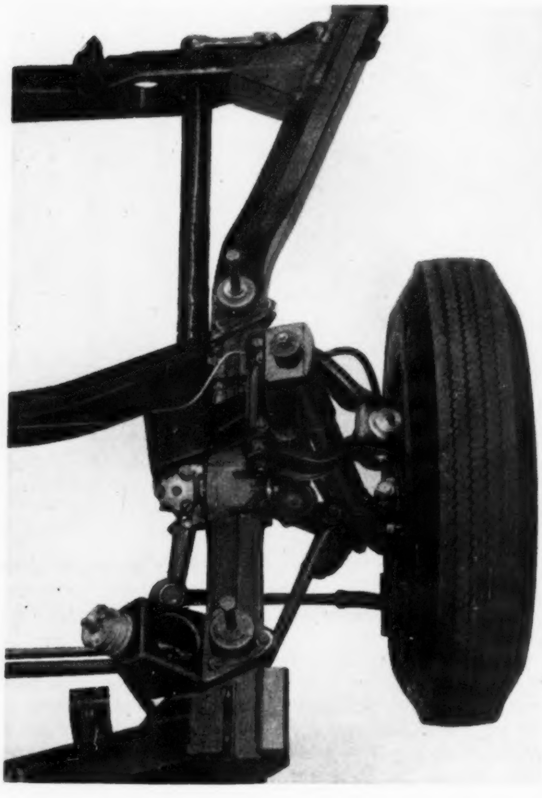
Pressed into the extremities of the radius arms are solid end-pieces of smaller diameter than the tubes; these end-pieces carry the flexible mountings. The rubber bush in the suspension link has an inner, bonded-on sleeve only; to provide for axial location, the hole in the link has chamfered ends, which are filled by the rubber when the bush is compressed by tightening the securing nut. The attachment of each radius arm to the sub-frame is by a sandwich type of end fitting, orthodox except for the method of providing adequate bearing area in the bracket. Local reinforcement increases the thickness of the bracket section round the hole, the diameter of which is considerably greater than that of the end-piece shank. On each side is a dished washer having a spigot that enters the hole. The rubbers are in the form of half-bobbins; they seat in the washers and fill the annular space between the shank and the spigots.

A square hole is broached in the boss of each suspension arm to accept the leading end of the torsion bar. This comprises five leaves of En.45 material, each having cross sectional dimensions of $1{\cdot}410 \times 0{\cdot}284$ in and an effective length of $31\frac{3}{8}$ in. Laminated construction was adopted to enable the bars to be kept short enough for their rear anchorage to be on the sub-frame, a feature that makes the front assembly virtually self-contained. Although the bars are of square section, they have to be fitted in a prescribed manner, to ensure that the correct basic angularity of the suspension arms can be obtained.

The angular rate of the spring is 16,550 lb-in/radian and,



Left: The adjusting arm on the rear end of each torsion bar is anchored by means of a draw-bolt and trunnion arrangement



Left: A view from above showing the front suspension layout and the method of mounting the upper wishbones. Above: The ball-joints of the suspension are well sealed, and large rubber bump stops are fitted

since the effective moment arm length is 12·437 in, the equivalent rate at the wheel is 107 lb/in. Attached on each side to the sub-frame is a rubber buffer of near-conical shape. It comes into contact with a dished abutment platform, bolted to the upper face of the link, relatively early in the upward travel of the wheel above the static load position. The buffer thus imparts a varying-rate effect to the suspension as well as acting as a resilient bump stop.

Angular adjustment of the rear anchorage of the bars is provided, to permit the accurate setting of the suspension on assembly. The rear portion of each bar passes through a hole in the third cross member of the sub-frame, immediately behind which it fits into the boss of the adjustment arm. This arm is pivot mounted by means of a shallow spigot on the front face of its boss, which registers in the hole in the reinforced rear wall of the cross member. Longitudinal location of the torsion bar is provided by a plain washer and an internal circlip fitted into a counterbore in the rear face of the boss.

The normal position of the arm is approximately horizontal, and its outboard end is forked and drilled to accept a trunnion bolt. This bolt is shouldered and its smaller-diameter portion passes through a vertical slot in the rear wall of the cross member, ahead of which it carries a plain washer and a lock-nut, for clamping purposes. The slot width is sufficient to accommodate the arcuate movement of the bolt. Adjustment of the arm position is effected by a draw-bolt, which passes through a threaded hole in the trunnion. The head of the bolt is at the bottom and has a spherical seating face that bears in a similarly shaped seating in a bracket on the cross member. By this means, the bolt is made self-aligning, to cope with the angular movement of the arm.

Because of the relatively narrow clamping face on the trunnion bolt, a second means is provided for locking the arm. It comprises a bolt carried in a short ear extending from the boss of the arm and passing through an arcuate slot in the cross member wall. As on the trunnion bolt, the assembly is completed by a plain washer and lock-nut within the cross member. Spanner access to these two clamping nuts is through holes in the member. A castellated nut, with a split pin, is fitted to the upper end of the draw-bolt, to prevent this from being inadvertently screwed right out of the trunnion during adjustment.

Like those of the rear suspension, the hydraulic dampers are of Woodhead Monroe manufacture, but they embody an auxiliary rubber bump stop. At the lower end of each is an attachment eye in which is a rubber bushed sleeve. This sleeve is clamped by a bolt between two pressed steel brackets; the brackets are bolted one on each side of the lower arm of the suspension. The piston rod of the damper is extended upward beyond the outer shroud and is connected by a rubber sandwich type end-fitting to the top flange of the large, fabricated bracket already described in the section on the sub-frame.

As stated earlier, an anti-roll bar is fitted to the front suspension. It has a diameter of ¾ in, and is installed ahead of the suspension assembly; its mid-portion is freely supported in rubber bushes attached to the underside of

the sub-frame side members. The swept-back ends terminate in eyes, to which the upper ends of two near-vertical rods are attached by means of rubber mountings of the sandwich type. The bottom of each rod is attached in a similar manner to an ear extending forward from the damper mounting bracket on the front face of the lower arm of the suspension.

The principal dimensions of the suspension system are as follows: castor angle, nil; camber angle, normally laden, +2 deg; camber angle at full bump, −1 deg; swivel angle, +4 deg; offset of swivel axis from centre of tyre contact area, $2\frac{9}{16}$ in; effective length of suspension upper links, 7·062 in; effective length of suspension lower links, 12·437 in; track change from static load to full bump, +1 in; roll centre height, $5\frac{1}{2}$ in. The periodicity is quoted as 76 c/min.

Steering

Since the steering layout of the Rover 3 Litre follows normal practice, little comment is necessary. The Burman recirculating-ball type steering box has a ratio of 17·6 : 1 at the straight-ahead position of the wheels, rising to 28·5 : 1 on full lock, and is attached by three bolts to the side member of the sub-frame. Its input shaft is splined and carries a two-arm spider, to which is bolted a rubber and fabric disc that forms a flexible coupling between the box and the steering column. The lower end of the column is fitted with a spider of similar design for attachment to the flexible disc.

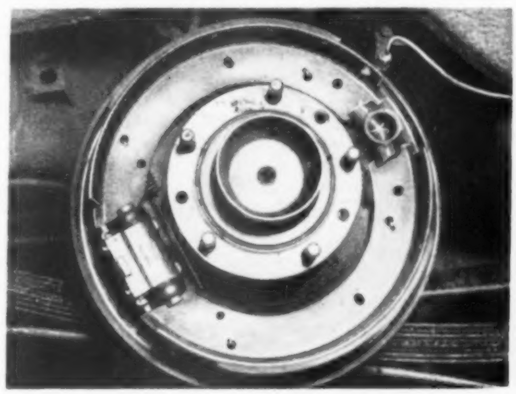
On cars with right-hand drive, the steering box is, of course, mounted on the right side member of the sub-frame, and the bearing housing for a slave lever is bolted to the other member, in a corresponding position. For left-hand drive, the disposition of the box and housing is reversed and these components, naturally, are then of the opposite hand. Two plain bearings of oil-retaining bronze support the slave lever spindle in its housing, the top cover of which encloses an adjustable friction damper that is splined to the upper end of the shaft. The steering box drop arm and the slave lever are both stampings, of 1035 steel, and are mounted on tapered splines on their respective spindles. They project forward of the spindles and, in plan view, are cranked in opposite directions, so that they first converge and then diverge; this shape is necessary to provide the correct geometry for the three-piece track rod.

Each lever has two bosses, one at the apex of the bend and the other at the forward end. These bosses are taper bored to receive the ball members of the linkage joints. The mid-portion of the three-piece track rod links the forward ends of the levers, while the outboard portions of the rod are connected to the intermediate bosses. From these, the latter portions trail appreciably to the forward-projecting steering arms, which are of En.18B steel and are attached by two bolts to the swivel members carrying the front wheel stub axles. Sealed ball joints, of Rover design, are fitted at the six articulation points of the steering linkage, which provides an Ackerman geometry.

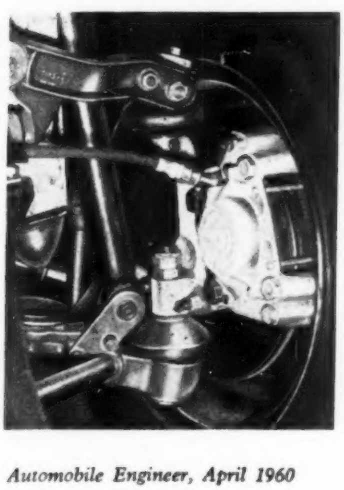
The nominal toe-in of the front wheels, in the straight-ahead position, is zero ± $\frac{1}{16}$ in, measured between the wheel rims at hub height. A turning circle of 40 ft has been obtained, which is satisfactory for a car of this size; $4\frac{1}{2}$ turns of the steering wheel are required from lock to lock. This latter figure may seem rather high for adequately quick response in an emergency, but the variable ratio of the box does, in fact, give sufficiently positive control near the straight-ahead position of the wheel; a relatively low overall ratio was necessary to obtain the traditional Rover lightness of control on a car weighing 32 cwt, of which 54 per cent is carried by the front wheels.

Wheels and brakes

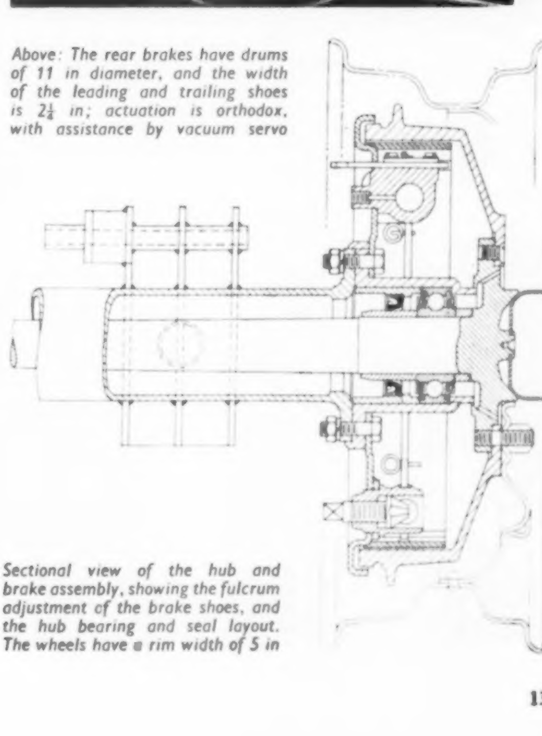
Pressed steel wheels, each with a five-stud attachment, are fitted. The rims have a diameter of 15 in and, as on the other models in the range, their width is 5 in; the use of these wide rims is claimed to give enhanced tyre stability and hence better cornering qualities. Tubeless tyres of



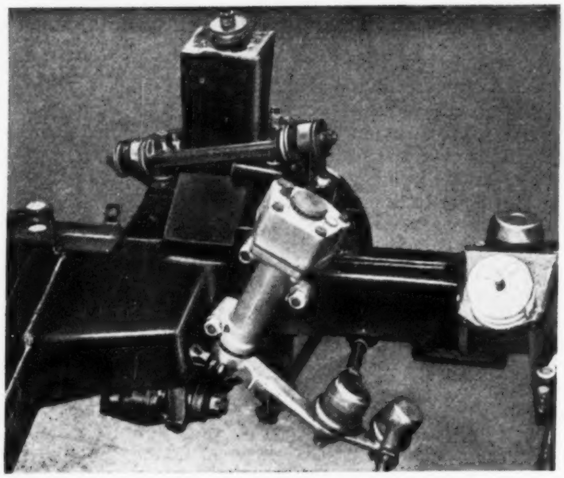
Above: The rear brakes have drums of 11 in diameter, and the width of the leading and trailing shoes is $2\frac{1}{4}$ in; actuation is orthodox, with assistance by vacuum servo



Left: Road dirt is kept from the Girling disc brakes by means of shrouds. The discs embody a device for preventing squealing



Sectional view of the hub and brake assembly, showing the fulcrum adjustment of the brake shoes, and the hub bearing and seal layout. The wheels have a rim width of 5 in



Each wishbone pivots on a forging attached to the sub-frame bracket; the suspension lower links have live pivot spindles. A friction damper is mounted on the upper end of the spindle of the steering slave lever

6·70 in section are standard equipment, but oversize, 7·10 in tyres can be fitted if required.

When the car was first introduced, Girling drum brakes were fitted at both front and rear. These brakes had drums of 11 in diameter, and the lining width was 3 in at the front and $2\frac{1}{4}$ in at the rear. Later, however, a change was made to Girling disc brakes for the front wheels.

The discs, which have a diameter of 10·794 in and a thickness of 0·567 in, are Chromidium iron castings. They embody an anti-noise device recently developed by Girling Ltd., which is said to eliminate squeal or grumble. This device is extremely simple: it comprises a ring of $\frac{9}{32}$ in diameter wire, the ends of which are welded together. The periphery of the disc is grooved to accommodate the ring which, however, has an inside diameter slightly larger than the diameter of the bottom of the groove. Difficulties were experienced during the development stages in evolving a method of ensuring the correct fit of the ring in the groove, but these have now been overcome. It is worthy of note that the device has been standardized also on the disc front brakes fitted to the Armstrong Siddeley Star Sapphire and Humber Super Snipe cars.

Each brake is equipped with a caliper of the fixed type, mounted behind the wheel axis and carrying two segmental friction pads, one of which operates on each side of the disc. Removal of the wheel is necessary before the pads can be changed. The material of the pads is Ferodo DS3 and their initial thickness is $\frac{1}{2}$ in, excluding the bonded-on steel backing plate. For each wheel, the friction and swept areas are respectively 13·7 in^2 and $128\frac{1}{2}$ in^2.

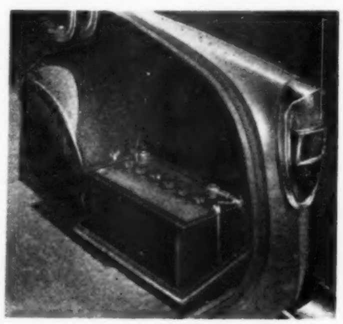
No change has been made to the rear brakes, which have leading and trailing shoes. The lining area per wheel is 38·9 in^2 and, for comparison with the front brakes, the corresponding swept area is 77·8 in^2. Ferodo AM2 lining material is employed. The Girling suspended-vacuum type servo unit, of 7 in diameter, is mounted on a bracket on the sub-frame, immediately behind the suspension assembly on the right-hand side. In order to avoid the risk of loss of braking efficiency in arduous conditions, or in the event of the engine stalling, the servo unit is connected to a vacuum reservoir. This is bolted to the inner valance of the right-hand front wing, ahead of the wheel, and is exhausted by the depression in the inlet manifold.

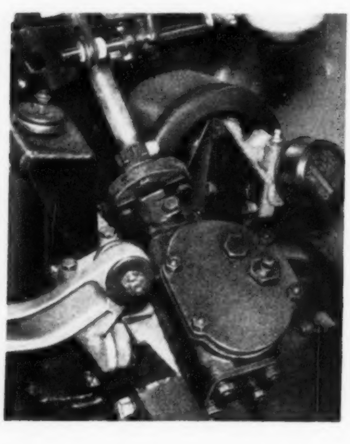
Mounted on the front face of the bulkhead behind the engine, the brake master cylinder is actuated directly by the pedal. A control valve on the servo unit proportions the degree of servo assistance to the effort applied to the pedal, thereby avoiding too violent a response to light application. The diameters of the slave cylinders in the brake units are such as to distribute 64·3 per cent of the braking effort to the front wheels and 35·7 per cent to the rear wheels. It is noteworthy that, with the drum brake system, the braking ratio was 66·8 : 33·2.

A handbrake lever, of the pistol grip type, is fitted below the facia panel. It is connected to a stranded steel cable that passes round two pulleys, the first with a vertical axis and the second with a horizontal axis. Beyond the second pulley, the cable runs downward to one arm of a bell-crank lever, transversely pivoted on a bracket attached to the underside of the toe board. To the other arm of the bell-crank is attached a longitudinal rod, which passes through one of the sleeved holes in the rear cross member of the sub-frame. At its rear end, this rod is connected to an arm on a cross shaft mounted on a transverse stiffener under the floor.

The cross shaft is linked by another rod to a slave lever, which is located near the axis of the front anchorages of the rear springs, in order to minimize road reaction if the hand brake should be used while the car is moving. Another rod

Right: The battery has a capacity of 57 amp-hr, and is stowed in the boot, on the right side





Far left: The Burman recirculating-ball steering gear is connected to the column by a flexible coupling of disc type. To the rear is the vacuum servo unit, which is of Girling manufacture. Left: Frontal lamps are neatly grouped; the sidelamps embody indicators visible to the driver

links the slave lever with a horizontal balance-lever carried on the axle casing; each end of the cross arm of the lever is coupled by a rod to one of the rear brakes. The geometry is so chosen that the pull on the rods is virtually perpendicular to the shoe plate. To obviate the need for periodic lubrication, the pivots of the brake pedal and hand brake system have oil-retaining bronze bushes.

Electrical equipment

All the electrical equipment, except the sparking plugs, is of Lucas manufacture. The 12-volt generator is a type C.45 PV6 unit, having a maximum continuous rating of 300 watts; it is belt driven from the crankshaft, in the normal manner, and its output is controlled by the usual current-voltage regulator. A combined vacuum and centrifugal automatic ignition-timing control is employed: it has a total range of 16 deg. The standard sparking plug equipment is the Champion N.5, but Lodge CLN-H plugs are installed if the lower compression ratio of 7·5:1 is specified. A type M.45G starter motor is fitted; the starter pinion has 11 teeth and there are 105 teeth on the flywheel rim. As already mentioned, the battery is housed in the boot; it has a capacity of 57 amp-hr at the 20 amp rate.

The side lamps are mounted at the front ends of the wing crowns, and they embody indicators to make them visible from the driver's seat. Below them are the separate amber direction indicators, of the flashing type; being of elongated shape and mounted on the front quarters, these indicators can be seen from the sides as well as the front. At the rear are the usual lamp units embodying tail and stop lamps, and direction indicators. For the home market, PL.700 headlamps are installed, and are equipped with 60/40 watt bulbs. The headlamp specification for the European market, however, is the F.700 type of lamp, together with 45/40 watt bulbs. It is rather surprising that the standard specification of a car of such high quality, and otherwise full equipment, should not include a built-in, matched pair of fog and long-range lamps.

Included in the standard equipment are windscreen washers, a reversing lamp and a cigar lighter. A practical feature of the facia lighting is that it illuminates the switches as well as the instruments. The interior lamps are switched on automatically when any door is opened but, in addition, each has its own finger-operated switch. Although it is not unusual for a luxury car to have a lamp illuminating the boot, it is of interest that on the Rover the automatic switch for this lamp is of the mercury type. Lucar connections are fitted throughout the wiring system of the car.

Lucas Alternator

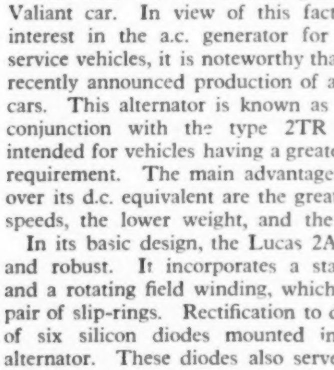
Preliminary Information Concerning a Robust and Compact Unit for Cars

REFERENCE was made last year, in the London Show Review issue of *Automobile Engineer*, to the adoption of an alternator for the electrical generation on the Plymouth Valiant car. In view of this fact, and of the increasing interest in the a.c. generator for commercial and public service vehicles, it is noteworthy that Joseph Lucas Ltd. has recently announced production of an alternator suitable for cars. This alternator is known as the model 2AC and, in conjunction with the type 2TR output-control unit, is intended for vehicles having a greater than average electrical requirement. The main advantages of the new equipment over its d.c. equivalent are the greater output at low engine speeds, the lower weight, and the absence of brush gear.

The Lucas alternator, type 2AC, which is only 5¾ in long and weighs 17¾ lb without a fan, is for use on cars having a high current requirement

In its basic design, the Lucas 2AC alternator is compact and robust. It incorporates a stationary output winding and a rotating field winding, which is energized through a pair of slip-rings. Rectification to d.c. is effected by means of six silicon diodes mounted in the end-plate of the alternator. These diodes also serve to prevent the reverse flow of current, so that a cut-out is not required. A built-in fan, of 6 in diameter, provides cooling air for the diodes and the machine itself.

The absence of a commutator permits the generator to be run at appreciably higher maximum speeds than could the corresponding d.c. unit. Consequently, the driving ratio can be chosen to give a useful output at speeds down to idling. The voltage is held between predetermined limits by a transistorized regulator in the control box. No external current-limiting device is necessary, since the reactance of the alternator is such that the current is restricted to 60 to 65 amps when the unit is cold and 52 to 57 amps when hot, at rotor speeds of up to 11,000 r.p.m. The nominal output, however, is 45 amps at 13·5 volts.



This is the type 2TR control unit, evolved specially for use with the 2AC alternator

According to the manufacturers, the 2AC alternator has a diameter of 6 in and a length of 5¾ in; its weight is 17¾ lb without the cooling fan, which weighs in the region of ½ lb. For comparison, a conventional d.c. machine for similar duties has an output of 35 amps, also at 13·5 volts, and a maximum speed of 9,000 r.p.m. It has a yoke diameter of 4·8 in, is approximately 8 in long and weighs 26½ lb without its fan.

In spite of the superior performance and lower weight of the alternator, an early changeover to a.c. generation on cars is not envisaged by Joseph Lucas Ltd. There are two reasons for this: the present d.c. equipment has been developed to a relatively high level and is generally satisfactory for the outputs normally required; also, its cost is lower, though the production of the a.c. equipment on a large enough scale should reduce the discrepancy.

Cary Laminaire Suspension

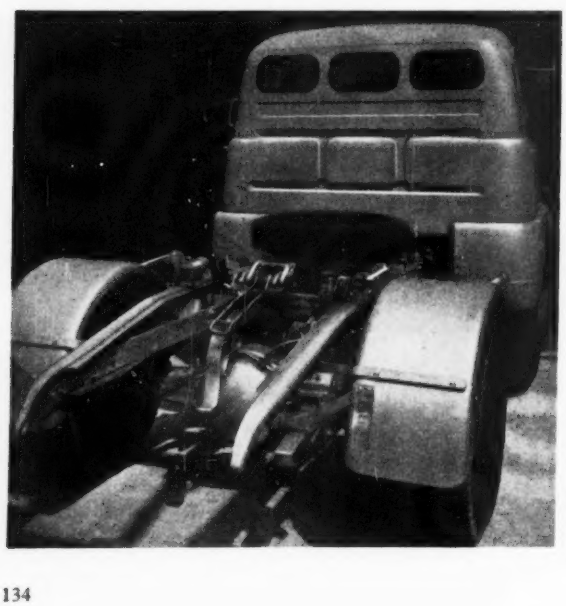
Ingenious Conversion System for Semi-Elliptic Springs, Providing a Progressive Rate Variation; Good Results Obtained on Commercial and Other Vehicles

FOR commercial vehicle applications, the conventional laminated semi-elliptic spring has the obvious disadvantage of a virtually constant rate. To avoid excessive deflection in the fully laden condition of the vehicle, the spring has to be excessively stiff for unladen running. Under the latter condition, not only does the driver have a rough ride but also the vehicle is subjected to severe impact loadings, with a consequent risk of the loosening of fastenings and even of fatigue failure of components; at the same time, wheel hop gives rise to an undesirably high rate of tyre wear. Even the use of normal helper springs is recognized as being only a palliative, since such springs do not cater for intermediate loadings of the vehicle. Also, the change of rate as the helper spring comes into action gives rise, in some instances, to sudden reactions.

Various attempts have been made throughout the years to solve the problem of combining good laden and unladen springing characteristics. In many respects, air suspension offers an excellent solution, but its use necessitates extensive redesigning, and the cost of the installation is considerably higher than that of leaf springs. It is therefore hardly surprising that commercial vehicle users displayed considerable interest in the Laminaire progressive suspension conversion, when it was introduced by William E. Cary Ltd., of Manchester, at the 1958 Commercial Vehicle Exhibition in London.

Since then, many commercial vehicles and trailers have been converted to Laminaire suspension, and several trailer manufacturers have adopted this system as original equipment. From the conversional aspect, the system has the advantages of relatively low cost and minimal time off the road. Few non-standard components are involved, and the vehicle need not normally be out of action for more than 24 hours.

This interesting Laminaire installation is on a Ford model D chassis, which has been modified for tractor work, and fitted with a Scammell coupling, by Merriworthy (Engineering) Ltd. The layout resembles that of the Albion tractor conversion, which is illustrated on another page



Laminaire springs were originally intended for rear axles of powered vehicles and for the axles of trailers and semi-trailers. So satisfactory have been the results obtained, however, that they are now being applied to front axles also. The range available caters for most vehicles having a pay load of $\frac{3}{4}$ ton or more. Those to which the springs have been fitted include passenger vehicles, among them a number of ambulances, and the improvement in their riding qualities is stated to have been very marked. It is noteworthy that the Austin Motor Co. Ltd. permits the installation of Laminaire springs without invalidation of the guarantee.

The Cary Laminaire spring is a development of the German Schomäcker design, for which the British company has a manufacturing licence. In its essentials, the layout consists of a special main spring, on the rear end of which bears a cantilever support spring, of much lower rate. The front end of the main spring fits the existing spring mounting bracket and pivot pin. For the pivot bearing, a normal eye is formed on the master leaf and, where space permits, the second leaf is curled round this eye, as a safeguard against failure.

Although the main spring is longer by as much as 25 per cent than the component it replaces, the longitudinal position of the axle relative to the vehicle is unchanged, since only the rear of the spring is lengthened. There is no direct attachment of this end of the spring to the chassis frame. Instead, the end is guided between side plates depending from a bracket bolted to the frame, in place of the standard shackle bracket. The replacement bracket, which also forms the anchorage for the cantilever support spring, is fabricated in box form, from mild steel plate. A rebound check is formed by contact between the down-curved end of one of the main spring leaves and a rubber roller mounted between the lower ends of the side plates.

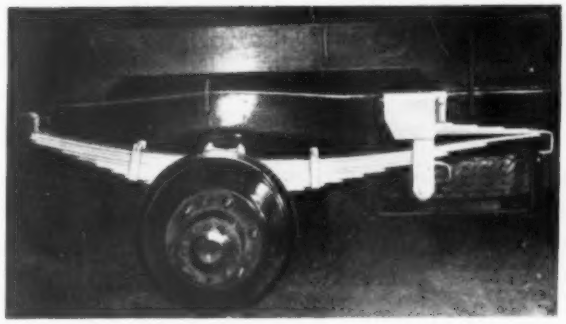
The support spring is rigidly mounted in a trailing position in the bracket, the side walls of which locate it laterally. Longitudinal location is effected by a dowel, formed on the lower end of the rivet holding the spring leaves together: this dowel registers in a hole in the lower wall of the bracket. The spring has a short, straight portion at its supported end, adjacent to which it is bent through a small angle, usually about 25 deg, and then extends straight to the rear. It is clamped in the bracket by a cast iron wedge, secured by two bolts passing through the front wall of the bracket. The lower, or clamping, face of the wedge has a ground finish to ensure accuracy of the profile, which follows the initial straight portion and the bend of the spring. However, the radius of curvature on the wedge is continued beyond the point at which the spring becomes straight, thus avoiding any heavy concentration of stress on the top leaf when the spring is deflected.

In the unladen state of the vehicle, the radiused tip of the

master leaf of the main spring contacts the corresponding leaf of the support spring, near its rear end, at an acute angle. Beyond the main spring, the end of the support spring master leaf is bent downward and forward through two right-angles. It thus forms a hook, the purpose of which is to limit longitudinal movement of the axle in the unlikely event of failure of the main spring eye. The ends of the remaining leaves have a truncated taper.

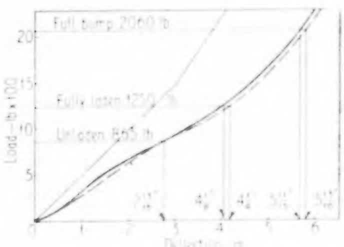
Under load, the behaviour of the two springs is as follows: their initial deflection, mainly of the softer support spring, closes the angle between them to zero. Then, as the load increases, the line of contact between the springs' adjacent leaves moves forward, but with a rolling and not a sliding action. Since this movement progressively reduces the effective length of both springs, it also results in a progressive increase in their combined rate. At full load, the line of contact intersects the centre-line of the support bracket.

Because this bracket occupies the position of the standard shackle bracket, and because the support spring then has zero effective length, the effective length of the main spring becomes the same as with the standard layout. It follows that, in these conditions, the suspension characteristics are unchanged from their original state, assuming the rate of the main spring rate not to have been altered. Moreover,

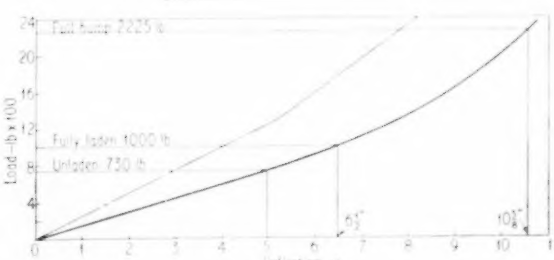


A typical Laminaire rear suspension conversion, on an Austin 2 to 3 ton chassis. The support spring and longer main spring are clearly visible, as is the vertical bracket that carries the rubber rebound roller

because of the initial low-rate deflection range of the support spring, the total available suspension travel is increased by a worthwhile amount relative to that of the normal suspension. A point worthy of comment is that, over the bump range beyond the fully laden level, the support spring is unstressed in bending, though it clearly is taking the reaction from the main spring in compression.



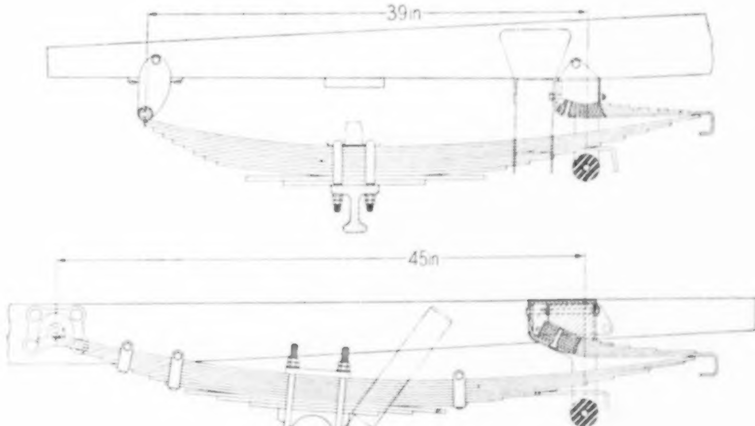
Load-deflection curves for the standard and Laminaire springs on the B.M.C. ambulance. On the left are the curves for the front suspension, and below are those for the rear springing



From the accompanying illustrations, it can be seen that there is a small gap between the main spring and the rebound roller. This does not mean that float, and hence chatter, can occur between the leaves. The springs have been drawn in the unladen condition of the vehicle, that is, with some initial deflection from the free position: on rebound the main spring contacts the roller before it separates from the support spring. Dampers are not regarded as essential for use with Laminaire suspension systems, because of the additional inter-leaf friction of the support springs, though they can be fitted if desired. Likewise, helper leaves of conventional design can be, and are, embodied in the main springs on applications where it is thought desirable to restrict sway or cornering roll.

The illustrations already mentioned have been selected as typical of a number of cases for which this form of suspension should be valuable. One pair shows the front and rear conversions for a B.M.C. ambulance. From the suspension designer's viewpoint, an ambulance is by no means an easy vehicle: although its loading variation is smaller than that of numerous other types, a smooth ride must be provided for the comfort of the sick or injured passengers, while at the same time, rapid travel is often

Reproductions of the general arrangement drawings of Laminaire conversions for the front and rear suspension systems of the B.M.C. ambulance. The support springs are anchored to their carrier brackets by a wedge, and the end of each bottom leaf is turned under the main spring, to limit the axle movement in the event of eye failure



essential, and this demands reasonable freedom from body roll during cornering.

The Laminaire springs evolved for this application provide a substantial reduction in the average rate over the range from the unladen to the fully laden conditions. At full load, they are still softer than the springs they replace, but at full bump the figures are approximately the same for the two systems, though the total deflection of the Cary springs is much the greater. As on the standard suspension, helper leaves are embodied to resist roll, and the rear suspension has telescopic dampers. The difference between the curves for the off-side and near-side springs results from the fact that the two have a different free camber, to compensate for the offset mounting of the engine. This point is, however, of no practical importance, since the discrepancy is most pronounced below the unladen position, and the rebound travel of the springs is relatively short.

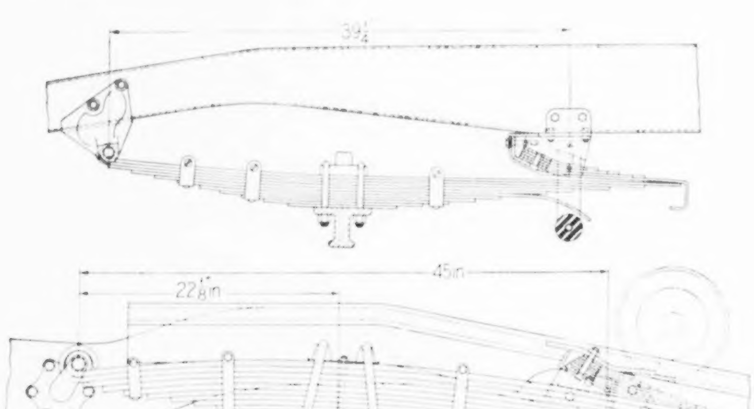
Of the second pair of drawings reproduced, the upper illustrates the conversion for the front suspension of an orthodox medium-weight commercial vehicle in quantity production—the 1958 Austin 2 to 3 ton model. With the vehicle unladen, the Laminaire system has a considerably lower rate than the standard spring, but in the fully laden condition the rate is higher, and it increases progressively to a total deflection 1⅝ in greater. Full bump load is consequently increased by no less than 1,250 lb.

The other example in this pair, the rear suspension of an Albion tractor unit, is noteworthy for the negative camber of the main spring, which has a double eye, and for the modified method of mounting the support spring. This mounting is necessitated by the presence of the guide rails of the coupling gear. The far greater light load deflection of the Laminaire suspension is clearly shown on the appropriate graph, and this difference must result in a marked improvement in the ride when the tractor is uncoupled or towing an unladen semi-trailer. However, over the bump range above the fully laden, static position, the rate is virtually the same as before. A somewhat similar Ford installation is shown in one of the half-tone illustrations. The chassis is basically a model D unit, but it has been shortened and fitted with a Scammell coupling, by Merriworthy (Engineering) Ltd., of Dartford, Kent. Because of the modifications to the chassis, it was necessary in this instance to drill additional holes for the mounting brackets.

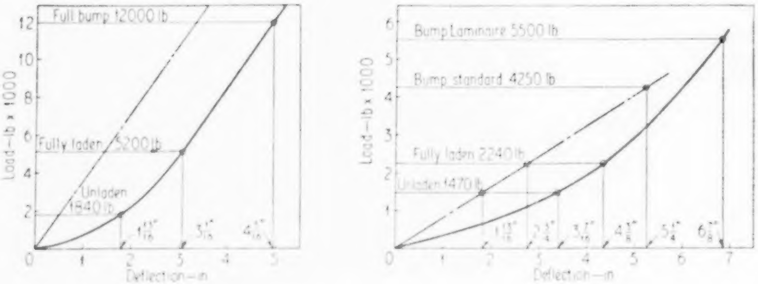
Trailers and semi-trailers present much the same problems as do tractor vehicles in respect of the rear suspension, though wheel hop in the unladen condition is undesirable because of its damaging effect rather than for any discomfort it may afford the crew. The satisfactory suspension of rigid or articulated tank vehicles, too, is notoriously difficult to achieve. Apart from the very high ratio of laden to unladen weight, and except for the articulated, frameless type, there is the purely structural matter of a rigid tank mounted on a relatively flexible frame. If the rear suspension beneath the tank is too stiff when the tank is empty, the resultant racking of the frame can cause fatigue failure of the tank mountings.

It is therefore apposite that the third pair of drawings should be of springs for the duties just mentioned. One application is to a Boden 14-ton semi-trailer and the other is to articulated petrol and bitumen tank units, manufactured by Thompson Bros. (Bilston) Ltd. A point of interest is that the Boden range of semi-trailers, comprising units for loads of 8, 10, 12 and 14 tons, was designed from the outset to have Laminaire suspension—for this reason, the appropriate graph includes no line for normal springing. The progressive stiffening of the suspension with load is again evident. No bump stops are fitted, the full dynamic load being taken by the main springs, which embody helper leaves.

The big reduction in rate in the unladen condition of the springs on the Thompson Bros. vehicles is obvious from the graphs, as is the fact that, at and above full load, the old and new rates are virtually the same. Bump stops were not initially fitted to this installation, but it was subsequently thought desirable to

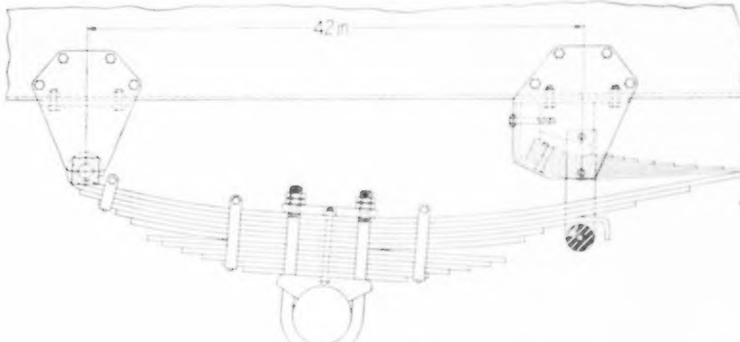


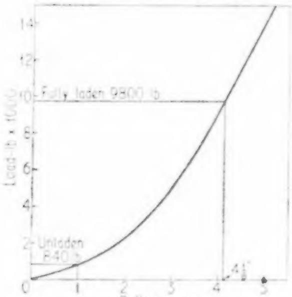
Left, above: The front spring conversion for the Austin 2 to 3 ton chassis. Left: A heavier rear suspension, fitted to an Albion tractor; a different mounting of the support spring is used in this case because of the siting of the guide rails

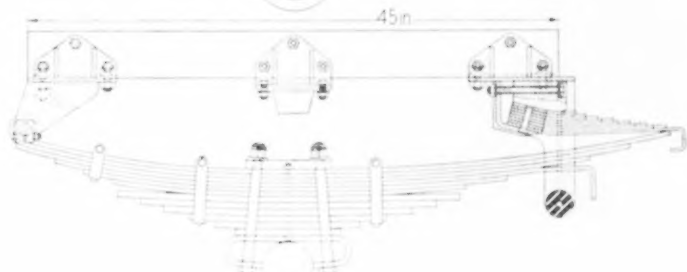


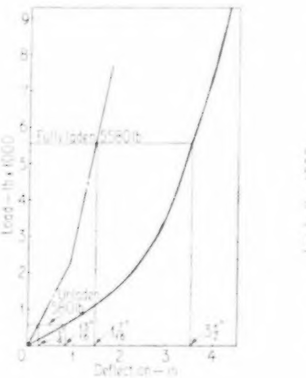
Load-deflection curves for the Austin and Albion springs illustrated above, and for their standard equivalents; the right-hand graph is that for the Austin springs. In both cases, the variation in the rate is pronounced, and the layout for the Austin provides a big increase in the bump load

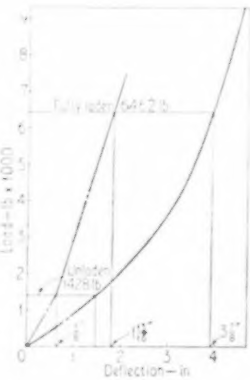
Right: A range of Boden semi-trailers is fitted as standard with Laminaire suspension; this example is for the 14-ton model. Right, below: Springs for Thompson Bros. articulated tank units for petrol and bitumen loads. Below are the appropriate curves for load against deflection. Those for the Thompson Bros. tankers are at the bottom; the right-hand graph is that for the heavier bitumen vehicle











make a positive restriction in the axle travel, since the tanks were intended for overseas service. To resist the rolling tendency, due to the high centre of gravity, the main springs of these tank units have an unusually large number of helper leaves.

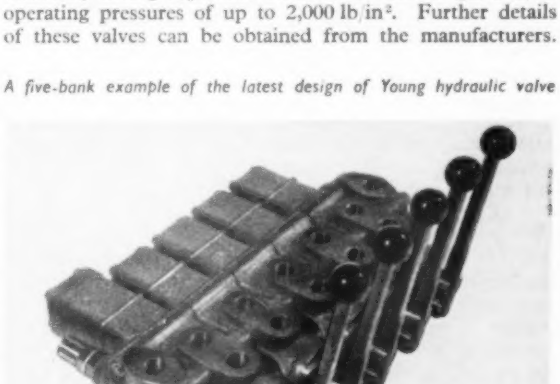
Strong evidence is provided by the various graphs that the Laminaire suspension does achieve its purpose of providing a progressively increasing spring rate. In this respect it can reasonably be claimed to compare well with an air suspension system. It admittedly lacks the latter's advantages of a constant platform height—of benefit mainly in passenger vehicle applications—and of the availability of the full bump travel of the suspension regardless of load. In compensation, however, it is less complicated, involves the minimum of alteration, and provides an increased total deflection relative to that given by orthodox springs.

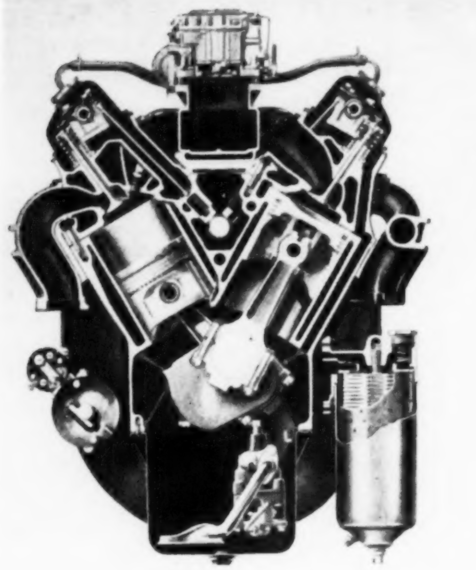
Multi-Bank Hydraulic Valve

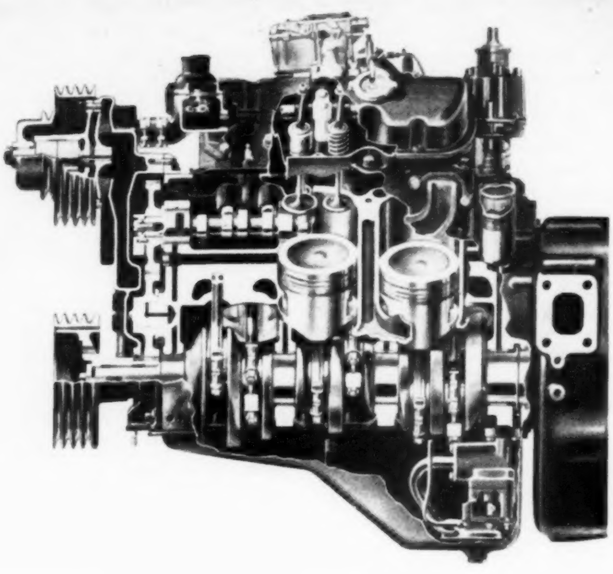
HYDRAULIC control systems are commonly employed on mechanical handling equipment, but one of the difficulties encountered is the tendency for mechanisms in use to be affected when others controlled by the same circuit are brought into operation. To overcome this problem, Joseph Young and Sons Ltd., of Kay Street, Bolton, Lancs, have designed and are producing a multi-bank valve assembly. The layout permits the use of any number of the valves in the bank, either individually or simultaneously, without any load drop, assuming that the cylinders controlled operate below the supply pressure.

This improvement has been effected by incorporating a check valve of special design in the valve body; the check valve is positioned in the pressure line between the valve spools. Modified production methods have been developed to ensure that the check valve and its associated porting have sufficient capacity to avoid the generation of heat under load. Any number of valves can be built up into a bank, using only three basic assemblies. The careful design of the spool profile and the hydraulic balance of the various valves in the bank are claimed to result in sensitive and accurate control. A range of sizes of units is available, having capacities between 15 and 45 gal/min, at operating pressures of up to 2,000 lb/in². Further details of these valves can be obtained from the manufacturers.

A five-bank example of the latest design of Young hydraulic valve







GENERAL MOTORS V-6 AND V-12 ENGINES

New Series of Spark Ignition Power Units for Use in Lorries

A SERIES of V-6 and V-12 petrol engines has recently been placed in production by the General Motors Truck and Coach Division, of Pontiac, Michigan, the largest American builder of buses and heavy lorries. Engines of these layouts are, of course, by no means new. The Packard Twin-Six, V-12 engine was used thirty-five years ago in some American luxury cars. General Motors built an experimental V-6 engine in 1950, but with 120 deg between the banks of cylinders. This engine was found expensive to produce and, on account of its extreme width, very difficult to install in the chassis. Both 60 deg and 65 deg V-type engines for ordinary and racing cars have been built by Ferrari and Lancia, while the Diesel Engine Division of General Motors, in Detroit, has had a V-6 automotive diesel engine in production for the past year.

Nevertheless, these new units represent the latest ideas in American lorry engine design. Also, the V-6 power unit has definite advantages for use in cars. It is, therefore, worth while to examine these engines in detail and to look into any special advantages they may possess.

Dimensions and performance

Given in the Table are the cylinder dimensions, the displacement and the power outputs of the engines in this new series. Although not listed separately, the model 305 engine is made up as three different versions; minor differences are introduced chiefly to limit the cost of engines for use in pick-up trucks and light lorries—in these applications the power units are not called upon to develop high outputs for long periods. The light-duty unit is referred to as the 305A. It is interesting to note that the twelve-cylinder engine has been considerably de-rated. It consists essentially of two model 351 units, placed end-to-end, and should therefore be capable of developing at least 360 b.h.p. at 3,400 r.p.m., but is rated at only 275 b.h.p.

In the case of the six-cylinder engines, the net power output, as installed and driving all accessories, is listed as from 84 to 88 per cent of the gross power output, and for the twelve-cylinder engine, it is 91 per cent of the gross output. The model 305 engines have a compression ratio of 7·55 : 1, while all the others have a compression ratio of 7·50 : 1. For all these units, the maximum gross torque corresponds to a b.m.e.p. of approximately 134 lb/in^2, and it is developed at a relatively low r.p.m.

Engine design

Fully machined combustion chambers are formed in the cylinder head. In shape, they are like a shallow basin, with pockets for the valves, and there is also a shallow spherical depression in the piston. These chambers have been developed, as the result of a long series of tests, to afford the lowest octane requirement for the compression ratio. Combustion chambers of this type, with almost vertical valves, are, of course, a compromise between the needs of minimum flame travel and an easy entry for the charge. Should a valve stick in the open position, it is still clear of the piston at its top dead centre station. The sparking plugs have been placed between the cylinder banks, away from the exhaust manifolds and close to the distributor.

The pistons are of permanent-mould cast aluminium alloy and are of the slipper type, cam ground and tin plated, to ensure uniform bearing on the cylinder walls and satisfactory running-in. All pistons have a steel expansion-control band, and, except those in the 305 engines, they also have a steel

insert in which the top piston-ring groove is machined. This insert greatly increases the life of the piston and top ring under severe operating conditions. The three compression rings are $\frac{3}{32}$ in wide, while the single oil-control ring is $\frac{3}{16}$ in wide.

Forged steel connecting-rods are employed and the one design is common to the whole range. Oil is supplied to each small-end bush by means of a hole drilled in the upper end of the rod. Both the big-end bearings and the main bearings are of M-400 material, supplied by the General Motors Moraine Products Division, or the F-77 material of the Cleveland Graphite Bronze Company.

Crankshaft and crankcase

On the model 305 units the crankshaft is of cast Armasteel, while those used in the other engines in the range are of forged chromium-alloy steel. Both the crankpins and the journals are hardened by the Tocco process. All the crankpins are $2\frac{13}{16}$ in diameter, while the main journals are $3\frac{1}{8}$ in diameter. The crankshafts for the six-cylinder engines weigh 110 lb. An interesting feature of the manufacture of the crankshaft is the use of a device to maintain an accurate radius and a perfect blend of the crankpin fillets between the pin and web. The grinding wheel is dressed continuously by a diamond tool, which traverses the wheel and swings around its corners while the grinding is in progress.

The crankshaft for the V-6 engines has four main journals and six separate crankpins which, as numbered from the front end, are spaced at 60 deg intervals from 1 to 2, 3 to 4 and from 5 to 6, and at 180 deg intervals from 2 to 3, 4 to 5 and from 6 to 1. A firing order of 1-6-5-4-3-2 has been adopted. Viewed from the front end, the bank on the left contains cylinders 2, 4 and 6, while that on the right contains cylinders 1, 3 and 5.

Counterweights, which occupy all the space available under the pistons, are placed on both of the end webs of the crankshaft. One large counterweight is placed on the central web of the shaft, that is, between numbers 3 and 4 crankpins. There are smaller counterweights on the rear web of number 2 throw and on the front one of number 5 throw.

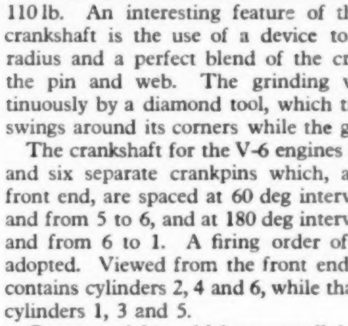
Except in respect of the position of the oil holes, the crankshaft used in the V-12 engine is to all intents a conventional shaft for a six-cylinder in-line engine. It has six fully counterweighted crankpins, on each of which is a pair of connecting rod big-end bearings.

The cylinder block and crankcase of the V-6 is a single nickel-chromium alloy iron casting, as also is that of the V-12 unit. Its sump face joint is 3 in lower than the crankshaft axis; this arrangement simplifies sealing problems and increases the crankcase stiffness relative to that of an engine with a joint at the level of the crankshaft axis. The main bearing caps are of alloy cast iron and are secured by two studs, except for the rear bearing cap, which has four. This latter cap fits snugly into a rectangular slot machined in the crankcase, in which it is sealed by means of cork inserts. The camshaft tunnel is placed conventionally between the arms of the V and is drained by drilled oil-holes, placed so as to retain a bath of oil into which the cams dip in the tunnel. Three oil galleries are drilled throughout the length of the block.

Cylinder head assembly

Cylinder heads of conventional design are employed. They are of chromium-nickel alloy cast iron, and are almost identical for all the engines of the range, four being used on the twelve-cylinder engines. Individual ports are provided for all valves, and the exhaust ports are fitted with pressed-in steel valve-seats. Six studs are arranged around each cylinder, there being fourteen studs per cylinder head. None of the stud bosses is tied to a cylinder barrel: this is to avoid distortion of the cylinder bores.

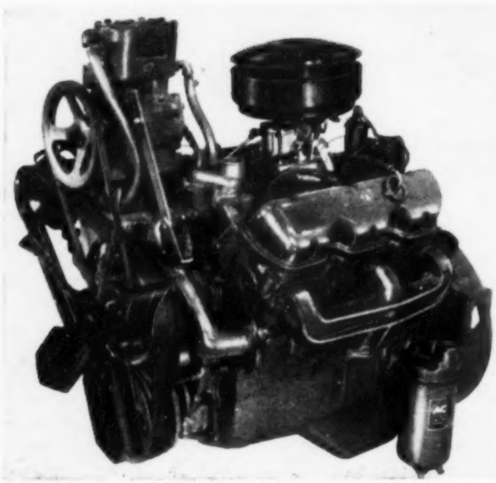
There is nothing new with regard to the design of the valve gear. For compactness, the single rocker shafts are set fairly close to the cylinder axes. Except in the 305A engine, all the inlet and exhaust valve springs seat on Thompson Products Rotocaps, which cause the valve to rotate slightly each time it is opened. This gives a substantial improvement in valve life. The tappets are of the barrel type and are slightly offset in relation to their cams, to encourage rotation and thus to distribute the wear over the whole tappet face. They can be withdrawn, for inspection, without removing the cylinder heads. Plain tappets are used on the six-cylinder engines and hydraulic self-adjusting tappets on the twelve-cylinder unit. On the latter it was

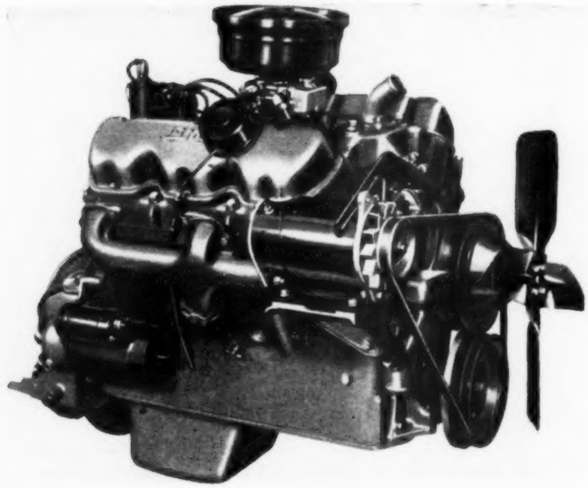


A *total flow through engine, with thermostats open;* **B** *flow through engine, with thermostats closed;* **C** *flow through radiator, thermostats open;* **D** *flow through by-pass, with the thermostats open*

Above: Curves showing the water flow characteristics. A large capacity pump is employed, since it must provide ample cooling for the twelve-cylinder as well as all of the smaller, six-cylinder versions

Below: The twelve-cylinder engine comprises basically two six-cylinder units but, of course, with a special crankcase and also a longer crankshaft



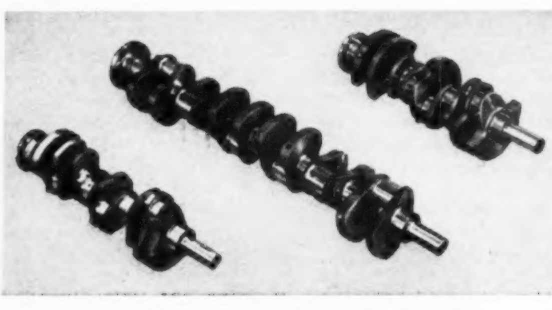


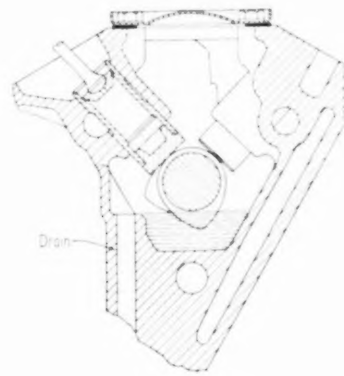
The two illustrations above, on the left and right respectively, are of the 401 and 305 engines. Both have the same stroke but their bores differ

Left: An unusually stiff, ribbed construction is employed for the exhaust manifold of the 401 unit, which develops 205 b.h.p. at 3,200 r.p.m.

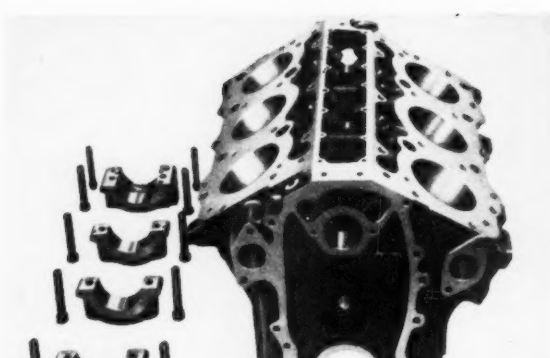
Below, left: A forged steel crankshaft is used in the 351 and 401 engines, while a cast Armasteel shaft is used in the 305 unit. Of the three shafts, the upper two are forgings, while the lower, on the left, is a casting

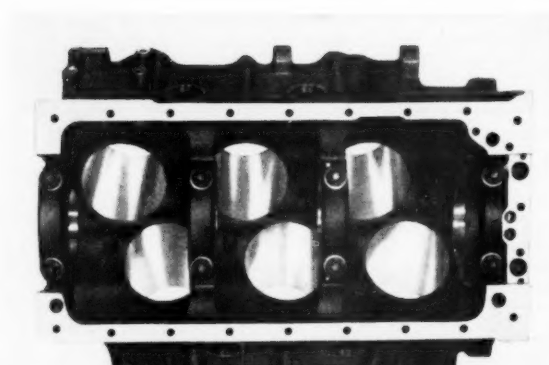
Below: The cams of the single central camshaft dip in an oil trough





Below: Two illustrations showing the arrangement of the crankcase and main journal bearing caps. The sump joint face is 3 in lower than the crankshaft axis, in the interests of stiffness and simplified sealing arrangement





felt that 24 clearances to adjust might lead to neglect, especially as, in certain installations, the rearmost adjustment devices would be difficult to reach.

So far as manifolding is concerned, the induction system incorporates the usual hot-spot, heated by gases passing through an auxiliary exhaust passage between the cylinder heads. The carburettor fitted to the 305 engines may be either a Holley unit, a Zenith 1½ in single-venturi unit, or a 1¼ in Stromberg Duplex, with two venturis, according to the type of service. The 1½ in Stromberg Duplex carburettors are used on the 702 engine. Velocity-type governors are employed on the smaller engines, while mechanical governors, described later, are used on the 401 and 702 units. The fuel pump is mounted high up on the engine front cover and is actuated by an eccentric bolted to the camshaft.

Cooling system

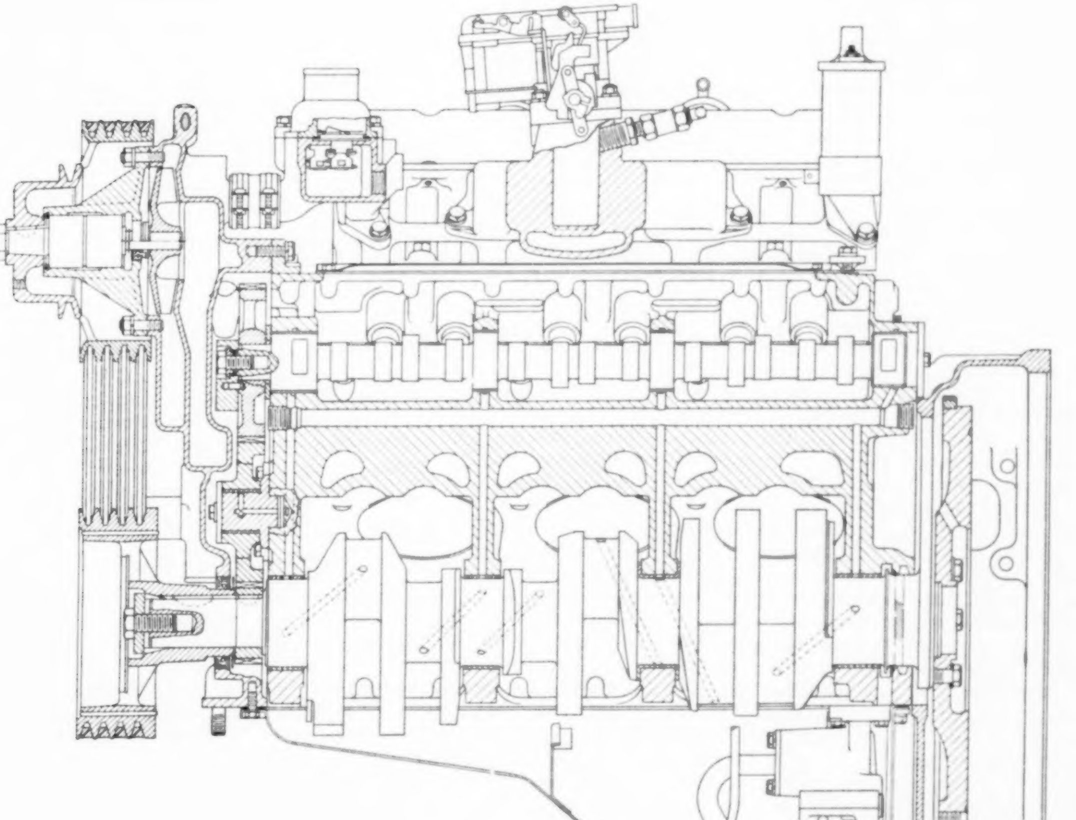
Common to the whole range, the water pump is exceptionally large, since it must provide ample cooling for the twelve-cylinder engine at its relatively low governed speed of 2,400 r.p.m. It is capable of circulating 150 gal/min (Imperial) through the cooling system. The water leaves the pump through two branch-pipes and flows to an opening in the front face of each of the two banks of cylinders. Thence it flows past the cylinder barrels and leaves the block through openings toward the rear of the top face. The layout of the engine—the crankshaft proportions, the bearing widths and so on—is such that ample space for the water-jacket core is left between the cylinders; this is so even in the 401 unit, which is the largest-bore engine of the range.

To cool the cylinder head, the water enters at the rear, and flows at high velocity towards the front, leaving through the thermostat housing. Either one or two thermostats are used on the 305 engines, according to the service; two are used on all the other six-cylinder engines, and three are fitted to the twelve-cylinder unit. All these thermostats are of the same size and type and are supplied by the Dole Thermostat Company.

Of special interest is the use of a large, open by-pass between the thermostat housing and the water pump intake. This ensures that local overheating, owing to lack of circulation during the warm-up period, cannot occur while the thermostat is closed. Even when the thermostat or thermostats are open, only half of the water flow goes through the radiator. At full load, the temperature rise through the engine is stated to be 5 deg F, while the temperature drop across the radiator is close to 10 deg F. For this reason, and because of the high rate of water flow, the engine is exceptionally uniformly cooled.

Lubrication system

A large oil pump, with a delivery of 12 gal/min (Imperial), is employed for the six-cylinder engines, and an even bigger pump delivers oil at 15 gal/min in the twelve-cylinder unit. In the galleries, the pressure is 60 lb/in^2. All the pumps are similar in design and have a four-tooth inner driving gear within and a five-tooth, idler ring-gear. They are each driven by a vertical shaft and spiral gears at the rear



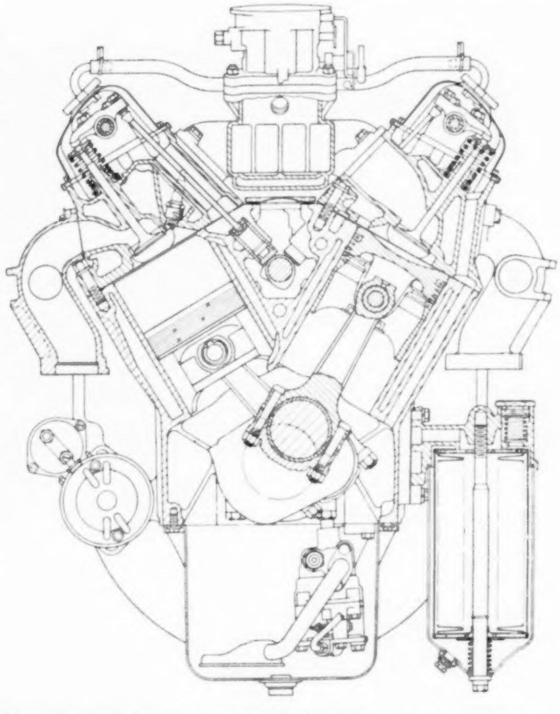
Longitudinal section of the engine, showing the layout of the lubrication system and the camshaft drive, which is effected by helical gears

Table—CYLINDER DIMENSIONS AND GROSS POWER OUTPUT OF THE ENGINES

Model	*Cylinders*	*Bore×Stroke*	*Displacement*	*Maximum gross output*
305	6 cyl.	4·25 in×3·58 in	304·7 in^3	165 b.h.p. at 3,800 r.p.m.
351	6 cyl.	4·56 in×3·58 in	351·2 in^3	180 b.h.p. at 3,400 r.p.m.
401	6 cyl.	4·87 in×3·58 in	400·9 in^3	205 b.h.p. at 3,200 r.p.m.
702	12 cyl.	4·56 in×3·58 in	702·4 in^3	275 b.h.p. at 2,400 r.p.m.

end of the camshaft. The pump is placed low enough in the oil sump to be self-priming, and it draws the oil through a large screened intake, placed close to the bottom of the sump. Before entering the galleries, the oil first passes through a full-flow filter, with a treated-paper element; this filter has a by-pass valve, to prevent stoppage of the oil flow should the element become clogged.

Attached to and driven from the lubricating oil pump in the 401 and 702 engines is the governor. This consists of a centrifugally-operated valve, which regulates the supply of oil to a pipe leading to a small diaphragm chamber on the carburettor. Normally, when the engine is running slower than its governed speed, the centrifugal valve is rotating at a rate at which it remains closed. As soon as the governed speed is reached, centrifugal force causes the valve to open, and oil from the lubrication system is admitted to the connecting pipe and thus to the diaphragm chamber. The diaphragm is pushed over and actuates a link to close the carburettor throttles. Although the driver's throttle control also moves the throttle, it does so through a spring lever device, so that he cannot over-rule the action of the governor. As the engine speed falls below the governed speed, the centrifugal valve closes and oil is released from the diaphragm cylinder. On the 702 engine, two carburettors are used and each has its own diaphragm cylinder.



Cross section of the engine; it would appear that the valve springs are of unusual form, presumably to give a rate that increases with opening

Crankcase ventilation

For crankcase and rocker cover ventilation, the 305 engine has a system such that the forward motion of the vehicle draws air and blows by-products out of the engine. Air is admitted to the engine through two small air-cleaners, one on each side of the cylinder-head covers. Having mixed with combustion gases that have found their way past the pistons and exhaust-valve stems, it is then drawn out through a tube that extends downward from the rear end of the crankcase.

A disadvantage of this system is that no air is circulated while the vehicle is standing, or moving slowly. In fact, because of convection effects, any air movement is in the opposite direction: in these circumstances, the air enters the engine through the tube and leaves by the air-cleaners.

The ventilation system used in all the other engines in the range is positive, and operates whenever the engine is running. Air enters the crankcase through a paper-element air cleaner and is drawn up to the cylinder-head covers and through hoses to a valve on the intake manifold. This is a double-acting poppet valve, which is spring-loaded to seal off the crankcase from the intake manifold while the engine is stopped; it also serves to prevent any flame from a backfire in the manifold from reaching the crankcase, where it might cause an explosion. Under full load conditions of operation, the valve is drawn from its seat and a free path is open for the air to pass from the crankcase into the intake manifold. At light load and idling, the valve is drawn right across its housing, to seat at the manifold end; in this position, only a small central hole is open to the manifold. In this way, the intake mixture ratio is not upset by ventilation air, nor is too large an amount of air drawn through the crankcase.

To sum up, the main advantage of the V-6 engine is its exceptional compactness. What has happened is that bearing materials are now so robust that it has become possible to reduce the overall length of engines, while still maintaining adequate space between the cylinders in a V-type unit. Engines of this type are easy to install in the chassis and access for repair or adjustment is good. The design of this series of engines has been worked out so as to afford a high degree of interchangeability of parts.

Repair by Welding

VALVE stem failure, more common in engines of racing cars than in those of other road vehicles, inevitably results in considerable internal damage, particularly if the cylinder head is of aluminium alloy. Because of the high cost of replacing a damaged light alloy head casting, it is common practice today to reclaim the damaged component. One of the most satisfactory methods is argon-arc welding, pioneered in this country by The British Oxygen Co. Ltd.

The valve seat inserts are first removed and then new metal is deposited, as necessary, by the argon-arc process. After the combustion chamber has been machined back to its original contours, new valve seats are shrunk into position. Further details of the argon-arc process and equipment are available from British Oxygen Gases Ltd., a subsidiary of The British Oxygen Co., St. James's, London, S.W.1.

Suspension Dynamics

Vehicle Vibration Analysed by a Stochastic Process

M. P. Bieniek, D.Sc. (Eng.)

This paper outlines the analysis of vehicle vibrations, based on the theory of stochastic processes. The application of statistical-probabilistic methods is justified by the essentially random character of the disturbances—roughness of the road—acting on the moving vehicle. The following parts of the problem are considered:
(a) The mathematical representation of random roughness of the road.
(b) The determination of dynamical characteristics—transfer functions, frequency response functions—of the vehicle.
(c) The analysis of vehicle vibrations from given characteristics of the road and dynamical characteristics of the vehicle.
(d) Methods of statistical determination of displacements, velocities, accelerations and stresses are given.

One of the serious difficulties in the theoretical analysis of vehicle vibration is the problem of the mathematical description of disturbances acting on the moving vehicle. These disturbances are caused by the roughness of the road surface.

In most papers on vehicle vibration, some special types of obstacles are assumed, for example, half sine waves, and the response of the vehicle to these specific disturbances is investigated. The results of any analysis are highly dependent on the arbitrarily assumed shape and dimensions of the obstacle and therefore their use is limited, even for the comparative study of vehicle suspensions.

New possibilities in the theory of vehicle vibration arise if the problem is treated as a stochastic process. The essentially random character of disturbances acting on the vehicle makes the stochastic methods the most appropriate for the solution of many important problems. Some advantages of the application of stochastic methods are:

1. The comparative study of different systems of suspension gives objective results, not influenced by arbitrariness in assuming specific shapes of the road surface
2. Objective information on the effectiveness of components of the suspension, such as springs and damping devices, becomes available
3. The mean values of the displacements, velocities and accelerations of the body, and probabilities or frequencies of their extreme values, may be determined. In relation to this, more rational criteria of comfort may be established and applied
4. The stochastic methods seem to be very promising in the evaluation of the stability and safety of the vehicle motion. To this class of problem belongs, for example, the following question: What is the probability that, or frequency with which, the wheel will lose contact with the road?
5. The application of stochastic methods to vehicle vibration study coincides very fortunately with recently developed methods of the analysis of the fatigue strength of metals

LIST OF SYMBOLS USED IN THIS ARTICLE

a	*distance from the centre of gravity to the front axle*
b	*distance from the centre of gravity to the rear axle*
c_f, c_r	*spring constants—forces producing unit deflections—of the front and rear axles, Fig. 3*
C_f, C_r	*spring constants of the front and rear springs, Fig. 3*
C_f', C_r'	*resulting spring constants of the front and rear springs and tyres, Fig. 4*
F	*general symbol for any magnitude, for example, displacement or velocity*
I	*moment of inertia of the body with respect to the lateral axis*
m_f, m_r	*mass of the front and rear axles and wheels, Fig. 3*
k_f, k_r	*damping coefficients of the tyres*
K_f, K_r	*damping coefficients of the dampers*
K_f', K_r'	*resulting damping coefficients of the dampers and tyres*
l	*wheelbase*
L	*total length of the measured profiles of the road, Fig. 2*
M	*mass of the body*
O	*centre of gravity of the body*
P	*dynamic force between tyre and road*
P_{st}	*static force between tyre and road*
$R(\tau)$	*correlation function*
S	*spring force*
T	$=L/v$
x, y, z	*coordinate system, Figs. 3 and 4*
s	*parameter of the Laplace transform*
v	*velocity of the vehicle*
Z	*vertical displacement of the centre of gravity of the body*
z_f, z_r	*vertical displacement of the centre of gravity of the front and rear axles and wheels*
z_{II}	*vertical displacement of the point II of the body, Figs. 3 and 4*
w	*ordinate of the road profile*
$\bar{w}(\omega)$	*Fourier transformation of $w(t)$*
$\widetilde{w}$	*time average*
w_f, w_r	*ordinates of the road under front and rear axles*
$\bar{F}(s), \bar{Z}(s), \bar{\phi}(s), \bar{z}_f(s), \ldots$	*transfer functions of $F, Z, \phi, z_f, \ldots$*
$\bar{F}(i\omega), \bar{Z}(i\omega), \bar{\phi}(i\omega), \bar{z}_f(i\omega), \ldots$	*frequency response functions of $F, Z, \phi, z_f, \ldots$*
$\dot{F}, \ddot{F}, \dot{Z}, \ddot{Z}, \dot{\phi}, \ddot{\phi}, \ldots$	*first and second time derivatives of $F, Z, \phi, \ldots$*
$\widetilde{F^2}, \widetilde{Z^2}, \widetilde{\phi^2}, \ldots$	*mean square deviations of $F, Z, \phi, \ldots$*
$\widetilde{\dot{F}^2}, \widetilde{\dot{Z}^2}, \widetilde{\dot{\phi}^2}, \ldots$	*mean square deviations of $\dot{F}, \dot{Z}, \dot{\phi}, \ldots$*
$\delta(t)$	*Dirac function*
σ^2	*mean square deviation*
τ	*measure of time interval*
ω	*parameter of the Fourier transform*
ϕ	*pitch angle of the body*
$\Phi_F(\omega)$	*power spectrum of any random variable denoted by index F*
θ	*integration variable*

under random stresses. This seems to be of importance for the strength analysis of the vehicle components.

Although the aim of this paper is at reviewing the general outlines of the analysis of vehicle vibration as a stochastic process it is not intended, or even possible, to discuss all the details of the subject. More elaborate treatment of the particular problems will follow this general introduction. The application of stochastic methods to similar problems is not new: they are extensively used in the analysis of communications systems, in the theory of control, the theory of flight and in other fields.

The mathematical analysis of the random variables used in this paper is based on the article by S. O. Rice[1]. Principles of the analysis of vibrations of a mechanical system may be found in the books by S. Timoshenko[2] and J. P. Den Hartog[3]. The application of the integral transforms is explained in the book by I. N. Sneddon[4]. A discussion of the response of a mechanical system to a random input is given in the book by H. S. Tsien[5]. Aspects of fatigue strength of metals, with

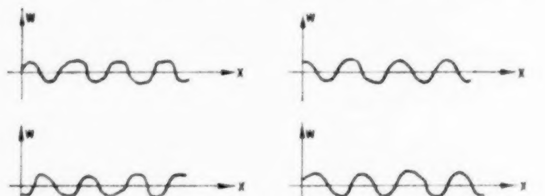
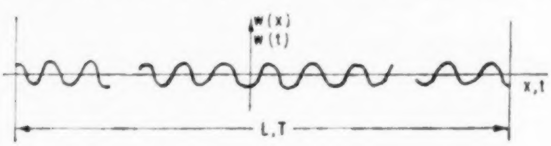


Fig. 1. Diagrammatic representation of profiles of a road, measured longitudinally along several tracks, for the preparation of Fig. 2

Fig. 2. Several measurements of the disturbances, applied by the road to the vehicle, combined to form one continuous representation for the analysis of the suspension characteristics of the vehicle



the application of probabilistic—statistical methods, is presented in an article by A. M. Freudenthal and E. J. Gumbel[6].

For the purpose of the analyses of the problems dealt with in this paper, the following assumptions will be made:

1. Disturbances acting on the vehicle are considered as stationary random
2. The vehicle is considered as a linear mechanical system, for example, spring forces are proportional to displacements and damping forces are proportional to velocities
3. Vibration in the vertical plane only will be considered and two models of the vehicle will be used: four-degrees-of-freedom system, shown in Fig. 3, and two-degrees-of-freedom system, shown in Fig. 4.

Analysis of the road surface

In the following discussion concerning some general principles of the mathematical analysis of the random roughness of the road surface, certain aspects requiring more careful treatment will be omitted by limiting consideration to one type of the road and sometimes by applying intuition instead of a rigorous argument. If measurements of the road surface are given in the form of profiles along several tracks selected at random, Fig. 1, and if the number of the tracks is large enough and each of them is of sufficient length, we may assume that all measured profiles, collected into one row, Fig. 2, provide sufficiently reliable statistical information concerning disturbances acting on a vehicle driven along that road.

Ordinates $w(x)$ may be measured from an arbitrarily assumed level. We will consider them as functions of time $w(t)$, introducing time by the relationship:

$$t = x/v \tag{1}$$

The following quantities and functions are used for the description of the properties of $w(t)$:

(a) The time average $\tilde{w}$:

$$\tilde{w} = \lim_{T\to\infty} \frac{1}{T}\int_{-T/2}^{+T/2} w(t)\,dt \tag{2}$$

It is possible to select the reference level such as to make $\tilde{w} = 0$, and this will be done in the following analysis.

(b) The mean square deviation σ^2:

$$\sigma^2 = \lim_{T\to\infty} \int_{-T/2}^{+T/2} [w^2 - (\tilde{w})^2]dt = \tilde{w}^2 - (\tilde{w})^2 = \tilde{w}^2 \tag{3}$$

(c) The correlation function $R(\tau)$ defined by

$$R(\tau) = \lim_{T\to\infty} \int_{-T/2}^{+T/2} w(t)\,w(t+\tau)dt \tag{4}$$

and having the following properties:

$$R(0) = \tilde{w}^2$$

If τ is large:

$$R(\tau) \to (\tilde{w})^2$$

(d) The power spectrum $\Phi_w(\omega)$, defined by:

$$\Phi_w(\omega) = \lim_{T\to\infty} \frac{1}{\pi T}\left|\bar{w}(\omega)\right|^2 = \lim_{T\to\infty} \frac{1}{\pi T}\bar{w}(\omega)\,\bar{w}(-\omega) \tag{5}$$

where $\bar{w}(\omega)$, is the Fourier transform of $w(t)$:

$$\bar{w}(\omega) = \int_{-\infty}^{+\infty} w(t)\,e^{-i\omega t}\,dt \tag{6}$$

There exist relationships between the correlation function and the power spectrum, known as Wiener-Khintchine relationships:

$$\Phi_w(\omega) = \frac{2}{\pi}\int_0^{\infty} R(\tau)\cos\omega\tau\,d\tau \tag{7}$$

$$R(\tau) = \int_0^{\infty} \Phi_w(\omega)\cos\omega\tau\,d\tau \tag{8}$$

In this analysis, principal use will be made of the power spectrum $\Phi_w(\omega)$

Basic solutions of the dynamical problem

In this section some dynamical characteristics of the system will be determined. Suppose that F is some magnitude being considered: it may be the displacement of some point of the body or axle; it may be the velocity or acceleration of some point; it may be the stress in some part of the body or suspension, and so on. The following three functions are useful for the determination of F under any road disturbances:

(a) Response to unit input, $F^*(t)$. This is the magnitude of F, changing in time, caused by a unit road disturbance, that is, a road disturbance given by:

$$w(t) = \delta(t) \tag{9}$$

where $\delta(t)$ is the Dirac function[4]

(b) Transfer function $\bar{F}(s)$. This is the Laplace transform of $F^*(t)$:

$$\bar{F}(s) = \int_0^{\infty} F^*(t)\,e^{-st}\,dt \tag{10}$$

(c) Frequency response function $\bar{F}(i\omega)$. This is related to $F^*(t)$ by the equation:

$$\bar{F}(i\omega) = \int_0^{\infty} F^*(t)\,e^{-i\omega t}\,dt \tag{11}$$

and may be obtained from $\bar{F}(s)$ by inserting $i\omega$ for s.

It is obvious that the above three functions for any magni-

tude F may be readily obtained if they are known for a displacement Z and rotation ϕ of the body, and displacements z_f and z_r of the front and rear axles, respectively.

Thus, it is necessary to determine response to unit input, transfer function, and frequency response function for Z, ϕ, z_f, and z_r. For this purpose, the vibration equations of the vehicle are considered. It is first done for the simplified system, Fig. 4, where:

$$z_f = w_f = w(t),\ z_r = w_r = w(t - l/v)$$

and only Z and ϕ are unknown. Two equations of motion are obtained by applying d'Alembert's principle to the body. Considering vertical forces we have:

$$-M\ddot{Z} - K_f'(\dot{Z} - \dot{\phi}a - \dot{w}_f) - K_r'(\dot{Z} + \dot{\phi}b - \dot{w}_r) - C_f'(Z - \phi a - w_f) - C_r'(Z + \phi b - w_r) = 0 \tag{12}$$

and, considering moments with respect to the centre of gravity of the body:

$$-I\ddot{\phi} + K_f'a(\dot{Z} - \dot{\phi}a - \dot{w}_f) - K_r'b(\dot{Z} + \dot{\phi}b - \dot{w}_r) + C_f'a(Z - \phi a - w_f) - C_r'b(Z + \phi b - w_r) = 0 \tag{13}$$

We now apply the Laplace transformation equations (12) and (13) and obtain:

$$[Ms^2 + (K_f' + K_r')s + (C_f' + C_r')]\bar{Z} + [(-K_f'a + K_r'b)s + (-C_f'a + C_r'b)]\bar{\phi} = (K_f's + C_f')\bar{w}_f + (K_r's + C_r')\bar{w}_r \tag{14}$$

$$[(-K_f'a + K_r'b)s + (-C_f'a + C_r'b)]\bar{Z} + [Is^2 + (K_f'a^2 + K_r'b^2)s + (C_f'a^2 + C_r'b^2)]\bar{\phi} = (-K_f'as - C_f'a)\bar{w}_f + (K_r'bs + C_r'b)\bar{w}_r \tag{15}$$

where Z, $\bar{\phi}$, $\bar{w}_f$ and $\bar{w}_r$ are Laplace transforms of Z, ϕ, w_f, and w_r, respectively, s being the transformation parameter. Note that equations (14) and (15) are linear algebraic equations with unknowns $\bar{Z}$ and $\bar{\phi}$. To obtain transfer functions $\bar{Z}(s)$ and $\bar{\phi}(s)$ we assume:

$$w_f = w(t) = \delta(t) \tag{16}$$

and, consequently:

$$w_r = w(t - l/v) = \delta(t - l/v) \tag{17}$$

and insert into (14) and (15) the Laplace transforms of w_f and w_r:

$$\bar{w}_f = 1,\ \bar{w}_r = e^{-sl/v} \tag{18}$$

By solving now equations (14) and (15) two transfer functions, $\bar{Z}(s)$ and $\bar{\phi}(s)$, are obtained; and by inserting $i\omega$ for s, frequency response functions $\bar{Z}(i\omega)$ and $\bar{\phi}(i\omega)$ are obtained. There is no need to write these functions explicitly here. In most cases, some approximate frequency response functions are sufficient. They are derived by making use of the fact that the expression

$$[(-K_f'a + K_r'b)i\omega + (-C_f'a + C_r'b)] \approx 0 \tag{19}$$

is very small, as compared to the other coefficients of equations (14) and (15). Sometimes it is zero, and approximations become exact solutions.

We have thus:

$$\bar{Z}(i\omega) = \frac{(C_f' + K_f'i\omega) + (C_r' + K_r'i\omega)e^{-i\omega l/v}}{-M\omega^2 + (C_f' + C_r') + (K_f' + K_r')i\omega} \tag{20}$$

$$\bar{\phi}(i\omega) = \frac{(-C_f'a - K_f'ai\omega) + (C_r'b + K_r'bi\omega)e^{-i\omega l/v}}{-I\omega^2 + (C_f'a^2 + C_r'b^2) + (K_f'a^2 + K_r'b^2)i\omega} \tag{21}$$

The first terms of the numerators of expressions (20) and (21) correspond to the front wheels passing the obstacle; the second terms correspond to the rear wheels passing the same obstacle.

Consider now the more rigorous system, shown in Fig. 3,

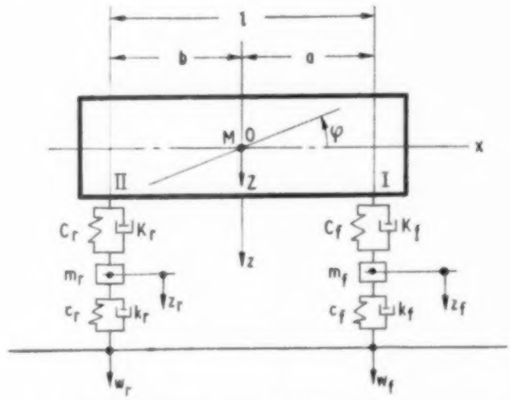


Fig. 3. Comprehensive diagram of the vehicle suspension system

where the masses of the axles and flexibility and damping of the tyres are taken into account. As has been mentioned, while the simplified system is good enough for determining the vibrations of the body, the more rigorous model is necessary for investigating such problems as: displacements of axles or wheel centres, dynamical forces on wheel bearings, pressures between tyres and road surface, the possibility of tyre losing contact with the road, and others.

The system has now four degrees of freedom and four equations of motion are to be derived. Again, make use of d'Alembert's principle and obtain:

(a) Considering vertical forces acting on the body:

$$\ddot{M}Z - C_f(Z - \phi a - z_f) + K_f(\dot{Z} - \dot{\phi}a - \dot{z}_f) + C_r(Z + \phi b - z_r) + K_r(\dot{Z} + \dot{\phi}b - z_r) = 0 \tag{22}$$

(b) Considering moments with respect to the centre of gravity of the body:

$$-I\ddot{\phi} + C_fa(Z - \phi a - z_f) + K_fa(\dot{Z} - \dot{\phi}a - \dot{z}_f) - C_rb(Z + \phi b - z_r) - K_rb(\dot{Z} + \dot{\phi}b - \dot{z}_r) = 0 \tag{23}$$

(c) Considering vertical forces acting on the front axle:

$$-m_f\ddot{z}_f - c_f(z_f - w_f) - k_f(\dot{z}_f - \dot{w}_f) + C_f(Z - \phi a - z_f) + K_f(\dot{Z} - \dot{\phi}a - \dot{z}_f) = 0 \tag{24}$$

(d) Considering vertical forces acting on the rear axle:

$$-m_r\ddot{z}_r - c_r(z_r - w_r) - k_r(\dot{z}_r - \dot{w}_r) + C_r(Z + \phi b - z_r) + K_r(\dot{Z} + \dot{\phi}b - \dot{z}_r) = 0 \tag{25}$$

By applying the Laplace transformation to equations (22), (23), (24) and (25), and inserting for $\bar{w}_f$ and $\bar{w}_r$ the expressions (18), it is possible to obtain four algebraic equations with four unknowns $\bar{Z}(s)$, $\bar{\phi}(s)$, $\bar{z}_f(s)$ and $\bar{z}_r(s)$, which are transfer

Table—EQUATIONS 26, FOR THE FOUR FREQUENCY RESPONSE FUNCTIONS

$\bar{Z}$	$\bar{\phi}$	$\bar{z}_f$	$\bar{z}_r$	**right side**
$Ms^2 + (K_f + K_r)s + (C_f + C_r)$	$(-K_fa + K_rb)s + (-C_fa + C_rb)$	$-K_fs - C_f$	$-K_rs - C_r$	0
$(-K_fa + K_rb)s + (-C_fa + C_rb)$	$Is^2 + (K_fa^2 + K_rb^2)s^2 + (C_fa^2 + C_rb^2)$	$K_fas + C_fa$	$-K_rbs - C_rb$	0
$-K_fs - C_f$	$K_fas + C_fa$	$m_fs^2 + (k_f + K_f)s + (c_f + C_f)$	0	$k_fs + C_f$
$-K_rs - C_r$	$-K_rbs - C_rb$	0	$m_rs^2 + (k_r + K_r)s + (c_r + C_r)$	$(k_rs + C_r)e^{-sl/v}$

functions of Z, ϕ, z_f and z_r respectively. Then, by substituting $i\omega$ for s, similar equations (26) are obtained for the four frequency response functions $\bar{Z}(i\omega)$, $\bar{\phi}(i\omega)$, $\bar{z}_f(i\omega)$, $\bar{z}_r(i\omega)$. These equations are shown in the Table. Their solution is rather complicated and leads to long expressions.

On the other hand it is not necessary to operate with such a high degree of accuracy. It seems that the following consideration leads to simplified solutions. So far as the vibration of the body is concerned, that is, $\bar{Z}(i\omega)$ and $\bar{\phi}(i\omega)$, the system from Fig. 4 should deliver almost the same

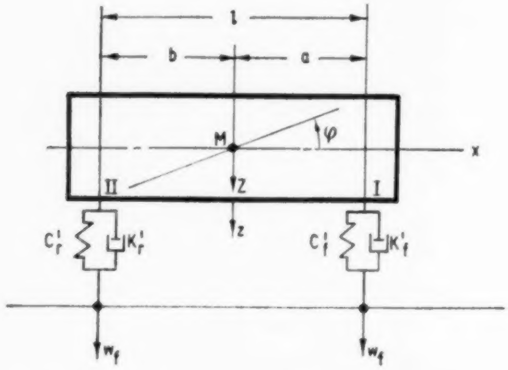


Fig 4. This simplified representation of a suspension system should give almost the same results, so far as vibration of the body is concerned, as the more complicated system that has been illustrated in Fig. 3

results as the system from Fig. 3; the coefficients in the Table lead us to expect that small inaccuracies in $\bar{Z}(i\omega)$ and $\bar{\phi}(i\omega)$ are of secondary effect on $\bar{z}_f(i\omega)$ and $\bar{z}_r(i\omega)$—sometimes even uniform motion of the body is assumed for the analysis of vibrations of the wheels and axles. Thus it can be assumed that expressions (20) and (21) also represent solutions of equations (26). By inserting these expressions into the third and fourth of equations (26) solutions for $\bar{z}_f(i\omega)$ and $\bar{z}_r(i\omega)$ are readily obtained.

$$\bar{z}_f(i\omega) = \frac{(C_f + K_f iw)\bar{Z}(i\omega) - (C_f a + K_f a i\omega)\bar{\phi}(i\omega) + (c_f + k_f i\omega)}{-m_f\omega^2 + c_f + C_f + (k_f + K_f)i\omega} \tag{27}$$

$$\bar{z}_r(i\omega) = \frac{(C_r + K_r i\omega)\bar{Z}(i\omega) + (C_r b + K_r b i\omega)\bar{\phi}(i\omega) + (C_r + k_r i\omega)e^{-i\omega l/v}}{-m_r\omega^2 + c_r + C_r + (k_r + K_r)i\omega} \tag{28}$$

Analysis of vehicle vibration

Given the characteristics of the road surface and the dynamical characteristics of the vehicle, it is possible to determine behaviour of the vehicle when it is moving at a specified velocity. Theoretically, it would be possible to solve the problem of vehicle vibration for $w(t)$, represented in Fig. 2, and obtain some diagrams representing displacements, accelerations, etc., as functions of time. This would give the most complete picture of the behaviour of the vehicle on the rough road. But it is obvious that such a procedure would require a tremendous amount of work, and therefore it is unrealistic when using classical methods of computation and is difficult when using computers; perhaps the easiest way would be by application of electrical analogue systems.

Much simpler, and satisfactory for practical purposes, is the determination of the power spectra and some properties of interesting components of the vibration. Given the frequency response function $\bar{F}(i\omega)$ of any magnitude, such as displacement, acceleration, on stress, it is possible to determine the power spectrum $\Phi_F(\omega)$ of this magnitude from the known relationship:

$$\Phi_F(\omega) = |\bar{F}(i\omega)|^2\Phi_u(\omega) \tag{29}$$

where $\Phi_u(\omega)$ is the power spectrum of the road disturbances. The derivation of relationship (29) has been given by Tsien[5].

For vertical and angular displacements of the body, and for vertical displacements of the axles:

$$\Phi_z(\omega) = |\bar{Z}(i\omega)|^2\Phi_u(\omega) \tag{30}$$

$$\Phi\phi(\omega) = |\bar{\phi}(i\omega)|^2\Phi_u(\omega) \tag{31}$$

$$\Phi_f(\omega) = |\bar{z}_f(i\omega)|^2\Phi_u(\omega) \tag{32}$$

$$\Phi_r(\omega) = |\bar{z}_r(i\omega)|^2\Phi_u(\omega) \tag{33}$$

where functions $\bar{Z}(i\omega)$, $\bar{\phi}(i\omega)$, $\bar{z}_f(i\omega)$ and $\bar{z}_r(i\omega)$ are given by (20), (21), (27) and (28) respectively.

Once the power spectrum for any given quantity F is determined, it is easy to obtain the mean square value $\widetilde{F^2}$ of this quantity:

$$\widetilde{F^2} = \int_0^\infty \Phi_F(\omega)\, d\omega \tag{34}$$

By using relation (34), the mean square values of Z, ϕ, z_f and z_r can be obtained.

It should be noted that the mean square value of a quantity gives important information. There exists a formula for calculating the average number of times N, per unit time, that the quantity F exceeds some given value F_c—or, more exactly, that $|F| > F_c$, F_c being positive—

$$N(F_c) = \frac{1}{\pi}\frac{\sqrt{\widetilde{\dot{F}^2}}}{\sqrt{\widetilde{F^2}}}\, e^{-F_c/2\widetilde{F^2}} \tag{35}$$

where $\widetilde{\dot{F}^2}$ is mean square value of $\dot{F}(t)$

$$\widetilde{\dot{F}^2} = \int_0^\infty \omega^2\Phi(\omega)d\omega \tag{36}$$

Consider a few particular applications of the general formula given in this section.

(a) First determine the power spectrum of the vertical motion of some point II of the body, just above the rear axle. The displacement of this point is Z_{II} and, from Fig. 3 or 4, it follows that:

$$z_{II}(t) = Z(t) + \phi(t)b \tag{37}$$

and consequently:

$$z_{II}(i\omega) = \bar{Z}(i\omega) + \bar{\phi}(i\omega)b \tag{38}$$

Using relationship (29):

$$\Phi_{zII}(\omega) = |\bar{z}_{II}(i\omega)|^2\Phi_u(\omega) = |\bar{Z}(i\omega) + \bar{\phi}(i\omega)b|^2\Phi_u(\omega) \tag{39}$$

which is the required power spectrum of z_{II}. It is also easy to determine the power spectrum of the vertical acceleration of point II.

From (37) follows:

$$\ddot{z}_{II}(t) = \ddot{Z}(t) + \ddot{\phi}(t)b \tag{40}$$

As is known, differentiation with respect to time results in the multiplication by s or $i\omega$ of the transfer or frequency response function, respectively. Thus, the frequency response function of the acceleration of point II is:

$$\bar{\ddot{z}}_{II}(i\omega) = -\omega^2\bar{z}_{II}(i\omega) = \omega^2[\bar{Z}(i\omega) + \bar{\phi}(i\omega)b] \tag{41}$$

and the power spectrum:

$$\Phi_{\ddot{z}II}(\omega) = \omega^2\Phi_{zII}(\omega) = \omega^4[\bar{Z}(i\omega) + \bar{\phi}(i\omega)b]^2\Phi_u(\omega) \tag{42}$$

(b) Consider now the dynamical force acting on the rear spring. By using the assumption that spring force and spring deformation are proportional, the relationship is obtained:

$$S(t) = C_r[Z(t) - z_r(t)] \tag{43}$$

S being positive when compressive.

From (43), come the frequency response function

$$\bar{S}(i\omega) = C_r[\bar{Z}(i\omega) - \bar{z}_r(i\omega)] \tag{44}$$

and the power spectrum

$$\Phi_s(\omega) = C_r^2|\bar{Z}(i\omega) - \bar{z}_r(i\omega)|^2\Phi_u(\omega) \tag{45}$$

which provide statistical information for the fatigue strength calculation of the spring.

(c) Consider now the dynamical force between the rear tyre and the road surface, Fig. 3:

It is equal to:

$$P(t) = c_r[z_r(t) - w_r(t)] \tag{46}$$

$P(t)$ being positive when compressive.

The frequency response function is then:

$$P(i\omega) = c_r[\bar{z}_r(i\omega) - \bar{w}_r(i\omega)] \tag{47}$$

where $\bar{w}_r(i\omega) = \int_0^\infty w_r(t - l/v)\, e^{-i\omega t}\, dt$

For the power spectrum it is possible to derive the expression:

$$\Phi_p(\omega) = c_r^2 \,|\bar{z}_r(i\omega) - \bar{w}_r(i\omega)|^2 \Phi_u(\omega) \tag{48}$$

Given $\Phi_p(\omega)$, it is possible to determine the mean dynamical contact force. Moreover, we are able to determine approximately, the frequency N_L of loss of contact between the tyre and the road. It is equal to half of the frequency with which the absolute value of the dynamical force is larger than the static force P_{st}. From (35):

$$N_L = \frac{1}{2\pi} \frac{\sqrt{\bar{\tilde{P}}^2}}{\sqrt{\tilde{P}^2}} e^{-P_{st}/2\tilde{P}^2} \tag{49}$$

where

$$\bar{\tilde{P}}^2 = \int_0^\infty \omega^2 \Phi_p(\omega) d\omega$$

$$\tilde{P}^2 = \int_0^\infty \Phi_p(\omega) d\omega$$

The factor 2 in the denominator of (49) arises from the fact that only the frequency of the inequality $P - P_{st} > 0$ or $P < -P_{st}$ is considered. In the expression (35), the frequency of $|F| > F_c$ or $F < -F_c$ and $F > F_c$ has been considered.

Conclusions

The described stochastic approach to the problem of vehicle vibrations is not more complicated than the classical methods. It offers perhaps the only realistic results describing the dynamical behaviour of the vehicle and the different factors influencing it. This treatment should be understood to be a first approximation: many topics require further and more refined investigation. There is no doubt that for best results, theoretical considerations should be completed by tests on models and analogue systems and by road tests.

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High-Speed Cinematography

THERE are numerous applications of high-speed cinematography in the automobile industry, and the technique has been used to solve many problems. However, the necessary equipment is expensive and requires expert handling. For smaller companies, or those whose need is only occasional, John Hadland and Co., Chipperfield, Herts, offer a full service on a daily contract basis. They are also able to give advice on equipment and technique to those about to use the method for the first time.

Spraying Heavily Pigmented Materials

A PROBLEM frequently met in spray finishing is that of the heavily pigmented material, which readily settles out when not being used. Most manufacturers who spray filler as part of their finishing process are familiar with the difficulties that can be encountered after work has been interrupted for only a few minutes. After leaving the pressure container, the heavy constituents settle out in the hose, and spraying cannot be resumed without a considerable waste of time and material.

Inexpensive equipment that solves this problem is now offered by Alfred Bullows and Sons Ltd., who originally evolved it for their own spraying work. A Bullows Pogo pump is used to feed the spraying material through a standard fluid hose to the spray gun, to which is attached a Y-piece. From the Y-piece, a second length of hose provides a return flow to the pump and hence to the container; the material in the container is kept agitated by this constant flow and return. This simple arrangement has now been in operation satisfactorily at the Bullows factory for over 18 months.

The Pogo pump can be used to draw the fluid from a mixing or storage tank, or it can be mounted on the existing pressure-feed container. It operates off the normal air line, using less than 3 ft^3/min of air at the relatively low pressure of 30 lb/in^2. An immediate application for this new development will be in the machine tool and similar industries, where finishing of castings is carried out on a large scale. The address of Alfred Bullows and Sons Ltd. is Long Street, Walsall, Staffs.

Girling Brake for Tractors

MANUFACTURE of the Ausco multiple-disc brake for tractors has recently been taken over by Girling Ltd. from Goodyear Tyre and Rubber Co. Ltd. The Girling version of the brake was first exhibited to the public in December at the Smithfield Show at Earls Court.. This disc brake is weatherproof and is equally efficient in each direction of wheel rotation. Although fully enclosed, for protection against mud and dirt, it has excellent fade resistance by virtue of the large frictional area of the eight rubbing surfaces.

An unusual feature of the brake is its method of actuation, which gives a servo effect. Pull on the operating rod causes relative rotation between two actuating discs in the brake assembly; since these discs are separated by a ring of steel balls housed in ramped pockets, the movement causes the balls to ride up the ramps, thus displacing the discs axially to apply the retarding load. According to the direction of wheel rotation, the angular travel of one or other of the discs is limited by a stop. The reaction of the braking force causes the free disc to tend to rotate with the wheel, thereby increasing the riding-up of the balls, and hence the retarding force. Excessive self-energization, which could readily lead to wheel locking, is prevented by careful choice of the ramp angle and of the lining coefficient of friction.

The Girling multiple disc brake will be made in a range of sizes sufficient for all normal tractor requirements. Its dimensions are very compact in relation to its energy dissipating ability. Details of the sizes and torque ranges that are available can be obtained from Girling Ltd., the address of whom is Kings Road, Tyseley, Birmingham, 11.

SPHEROIDAL GRAPHITE CAST IRON

Its Properties and its Applications in Automobile Engineering

G. FitzGerald-Lee

In ordinary cast iron, carbon occurs in the form of graphite flakes, and most of these flakes are connected to one another between the particles of iron, thus causing discontinuities in the material and raising stresses, which makes it weak and brittle. Although the isolated type of graphite in malleable iron is more compact and gives higher mechanical strength and ductility, this condition can be attained only by extended heat-treatment, which is generally limited to light sections. The Mond Nickel Company Ltd. is now producing and supplying a cast iron in which the graphite is in the form of spheroids, resulting from the presence of a small amount of magnesium retained in the iron. In this metal, S.G. (spheroidal graphite) iron—referred to in this article by its rather more apt alternative name of ductile iron—there are no discontinuities, thus the properties of the metal, especially toughness, are considerably better than those of its other ferrous predecessors, and there is no limit to the size and nature of the castings that can be produced.

Perhaps the most practical way of describing the properties of this remarkable cast iron is to mention the fact that a solid, round bar of $1\frac{3}{16}$ in diameter can be bent around a radius of $1\frac{3}{4}$ in—only about $\frac{1}{2}$ in more than its own diameter. In the $\frac{1}{2}$ in $\times \frac{1}{8}$ in flat bar form it can be twisted through two complete revolutions along its length or bent cold around a radius of $\frac{3}{8}$ in; in none of these conditions is there any sign of fracture. Although a true cast iron, the material combines high strength, toughness and ductility with excellent casting properties and good machinability; its modulus of elasticity is high, and it is resistant to corrosion and abrasion. It is in many ways superior to grey cast iron, malleable iron, non-ferrous alloys, and cast or forged steel. There are five types of ductile iron; they are pearlitic, hardened, ferritic, alloyed, and austenitic. These may be regrouped into eight grades: as-cast, annealed, annealed-high ductility, as-cast and stress relieved, normalized, quenched and tempered, surface hardened, and austenitic, for example, S.G. Ni-Resist cast iron, produced by the same firm. The annealed grade is the most widely applicable to automobile engineering, while austenitic is the least, its main uses being in power engineering. In Table 1, the mechanical properties of ductile iron are compared with those of the normal high-duty flake-graphite cast iron.

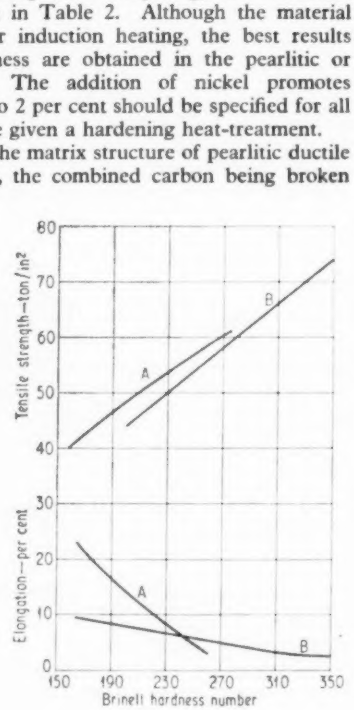
Table 1—MECHANICAL PROPERTIES OF S.G. IRON, AS CAST AND ANNEALED, COMPARED WITH THOSE OF HIGH-DUTY FLAKE-GRAPHITE CAST IRON

	High-duty, flake-graphite cast iron	*S.G. iron*	
		As cast	Annealed
Maximum stress, ton/in²	18–22	35–45	24–35
Yield point, ton/in²	—	25–35	17–25
Elongation, per cent	nil	1–5	10–25
Transverse rupture stress, ton/in²	38–42	55–65	55–60
Compressive strength, ton/in²	60–65	65–80	48–58
Compressive yield strength, ton/in²	—	32–40	24–32
Elastic modulus, $\times 10^6$ lb/in²	18	25	25
Brinell hardness	210–240	230–280	140–180
Impact Izod, 10 mm², notched, ft-lb	1	4	12
Impact Charpy, 10 mm², unnotched, 40 mm span, ft-lb	2	8	70
Endurance limit, unnotched, ton/in²	8·5	13–18	11–13
Endurance limit, notched 0·05 in rad., ton/in²	8	9·5	8·2
Endurance ratio, unnotched	0·4	0·46	0·45
Damping capacity ratio	4	2	1

Ordinary as-cast sections, up to 2 in in thickness, in which the carbon is mainly combined and the structure is of ferrite mixed with cementite and spheroids of graphite, are made in pearlitic ductile iron, a hard metal with twice the strength of grey cast iron. Its properties can be improved by normalizing, and the stresses in complex castings and in parts requiring maximum stability of dimensions are relieved by heat-treatment. Ductile iron can be hardened by quenching and tempering; the resultant properties are shown in Table 2. Although the material responds to flame or induction heating, the best results and maximum hardness are obtained in the pearlitic or normalized grades. The addition of nickel promotes hardenability, and 1 to 2 per cent should be specified for all castings that are to be given a hardening heat-treatment.

During annealing, the matrix structure of pearlitic ductile iron becomes ferritic, the combined carbon being broken down and distributed as graphite spheroids, the resultant material having maximum toughness and ductility with increased elongation. The presence of nickel makes annealed ductile iron stronger, without appreciably affecting its impact value or elongation. Adding 1 per cent of nickel results in a single metal which has a high strength and also elongation and shock-resistance. During the slow cooling that follows solidification, heavy sections are partly annealed and usually have, as-cast, a structure of mixed pearlite and ferrite.

Curves showing the relationship between the elongation, tensile strength and the hardness of ductile iron



Special matrix structures for acicular, austenitic or martensitic castings—properties of which are also shown in Table 2—are developed by alloying certain amounts of nickel, molybdenum, chromium or other elements. Austenitic ductile irons contain up to 30 per cent of nickel, resulting in a material with good mechanical properties and corrosion-resistance; other austenitic grades, of the nickel-manganese type, are used for their non-magnetic properties

Table 2—TYPICAL PROPERTIES OF HEAT-TREATED AND ALLOYED S.G. IRON

Type	*Treatment*	*Tensile strength ton/in²*	*0·5 per cent proof stress ton/in²*	*Elongation per cent*	*Brinell hardness*	*Izod impact, 10 mm², notched, ft-lb*
S.G. iron containing about 1 per cent Ni, as cast		35–44·5	25·5–35	1–3	220–280	0·7–1·5
Stress-relieved	2 hr at 550 deg C	38–47·5	25·5–35	3–5	220–270	2
Normalized	2 hr at 850 deg C, air cool	44·5–57	32–44·5	3–7	240–300	5
	Quenched from 850 deg C, tempered at 350 deg C	57–66·5	47·5–60·5	1	450–525	0·7
Oil-quenched and tempered	Quenched from 850 deg C, tempered at 500 deg C	60·5–70	51–60·5	1–3	350–450	3·5
	Quenched from 850 deg C, tempered at 600 deg C	47·5–57	38–47·5	3–5	250–325	5
Flame-hardened	Surface-hardened	—	—	—	600–750	—
Annealed	2-4 hr at 850-900 deg C, 6-12 hr at 700 deg C	25·5–30·5	17–22·25	12–25	140–180	7–15
Acicular S.G. iron, about 3 per cent Ni and about 1 per cent Mo		47·5–63·5	38–51	3	280–350	6
Austenitic S.G. iron, 22 per cent Ni		25·5–30·5	14–18	10–25	140–200	15

as well as for their strength. In acicular irons containing nickel and molybdenum, the strength of the material is equal to that of high-tensile steel and, when it is normalized or quenched and tempered, even higher strengths can be obtained. The stress-strain curve of ductile iron resembles that of steel rather than that of cast iron, because the strain is directly related to the stress up to a definite yield point, from which the proof stress can be accurately ascertained. This yield point—especially in ferritic ductile iron with 2 per cent of nickel—is well above that of malleable iron, thus the material can withstand heavy loads without permanent deformation.

Taken together with its good yield strength and elongation, the modulus of elasticity of ductile iron shows that the metal is stiff and with high resistance to permanent deformation, yet it withstands a considerable amount of bending and twisting without appreciable loss of strength. As section thickness increases, so the mechanical properties of cast metals usually decrease, which is also the case with ductile iron, except in that the rate of decrease is very much less than with flake-graphite cast irons. This characteristic, together with its generally good mechanical properties, are the reasons why ductile iron, in massive sections, has three times the strength of ordinary cast iron of similar sections: the tensile strength of 40-inch sections is about 20 ton/in².

Annealed, ferritic ductile iron has twelve times the shock-resistance of ordinary cast iron, other grades also being superior but to a lesser degree. So, for impact-resisting castings, the fully annealed, ferritic grade—which also has an elongation of up to 25 per cent—should be used. As with other high-strength ferrous metals, the ratio of limiting fatigue strength to tensile strength, that is, the endurance ratio, falls with increasing tensile strength, but the fatigue strength of ductile iron is initially much higher than that of even straight carbon steel. Table 1 also shows that the effect of sharp notches, as indicated by the stress concentration factor, is much less than with steel, so that the working fatigue strength of ductile iron is decidedly high, making it especially suitable for such parts as crankshafts.

In the softest condition of the metal, full annealing causes carbon to leave the matrix for the spheroids, to form ductile ferrite with a Brinell hardness of 150. Soaking for a short time at austenizing temperature causes the carbon to enter the matrix, resulting, after quenching, in a Brinell hardness of 700—this is much higher than the figure for white cast iron and twice that for mottled cast iron. Flame or induction treatment can produce similar hardness locally. Hardness figures between 150 and 700 Brinell can be obtained by various oil-quenching and tempering treatments. Normalizing results in good mechanical properties, with a Brinell hardness of about 280. The relationship between hardness, elongation and tensile strength is shown in the accompanying graph, which is reproduced opposite.

Table 3—PROPERTIES OF S.G. IRON AT HIGH TEMPERATURES

Properties	*As cast*	*Annealed*
Creep strength (1 per cent in 10,000 hr) at 427 deg C	5–11	7–12
Stress rupture: 100 hr at 427 deg C	18·0–24·0	13·5–15·5
1,000 hr at 427 deg C	13·5–18·0	11–12
100 hr at 650 deg C	1·7–2·1	1·5
1,000 hr at 650 deg C	1·1–1·3	0·9–1·0
Melting range, deg C	1,120–1,180	
Thermal expansion (mean values × 10^{-6} per deg C), over the temperature range:		
20 to 100 deg C	11·5	
,, ,, 200 deg C	12·2	
,, ,, 300 deg C	12·6	
,, ,, 400 deg C	13·1	
,, ,, 500 deg C	13·3	
,, ,, 600 deg C	13·5	

Ductile iron, in common with most cast irons, is highly resistant to abrasion; it also resists unequally distributed loads and those involving impact, such as may be experienced in heavy-duty gear wheels, in which the material has to withstand surface breakdown or tooth breakage as a result of fluctuations in load or of impact. Even when the lubrication fails, the metal is resistant to pitting, scuffing or galling, which in other metals could well result from working dry. Nickel-alloyed heat-treated grades are the best for gears. The presence of graphite increases the damping capacity of a cast iron, giving ductile iron an advantage over steel in absorbing vibration.

The structure of flake-graphite cast iron changes at high temperature and the metal suffers gas penetration, with resultant internal oxidation and, therefore, growth. Graphite spheroids are, however, isolated from one another and do not form continuous paths for the penetration of gases, as do graphite flake networks. Thus, ductile iron is more dimensionally stable at high temperatures and, therefore, resists growth better than ordinary cast iron; surface oxidation is also much less, and high alloy austenitic ductile iron is very resistant to heat. As may be seen from its creep strength, shown in Table 3, ductile iron is also very suitable for medium-temperature work, several of the more highly alloyed grades having creep strengths equal to those of some alloy steels. These high alloys are very resistant to thermal shock—unlike ordinary cast irons—as can be appreciated from the following experiment. A 6 in flanged gate valve of ductile iron was bolted to a steel pipe and heated in an oil fire; when it was red hot, at 730 deg C, it was quickly quenched with a fire hose. Subsequent inspection revealed that there was no damage to the valve, although it was tested to three times its rated pressure; the stresses to which it had been subjected, however, were so severe that the steel fittings were buckled and the steel bolts securing the valve to the pipe were sheared off.

As does that of other metals, the impact-resistance of ductile iron decreases with temperature, but its initial shock-resistance, and also its resistance to low-temperature

Table 4—SUMMARY OF FATIGUE PROPERTIES OF S.G. IRON AND OTHER MATERIALS; WOHLER TESTS

	Tensile strength, ton/in²	*Fatigue strength, ton/in²*		*Endurance ratio*	*Stress concentration factor* notch sensitivity
		unnotched	notched	unnotched	
Flake-graphite grey cast iron	20·0	8·0	6·0	0·40	1·3
S.G. iron, as cast	40·0	18·5	10·5	0·46	1·8
S.G. iron, annealed	30·0	13·5	13·2	0·45	1·6
S.G. iron, quenched from 900 deg C, tempered at 600 deg C	60·5	22·0	13·5	0·37	1·6
Forged carbon steel	35·0	16·0	8·0	0·45	2·0

embrittlement, are both much greater than those of ordinary cast irons: for resisting embrittlement it is even better than wrought mild steel. Composition largely governs low-temperature properties; annealed, high-ductility ductile iron can be used at temperatures down to 0 deg C, but its properties can be maintained at even lower temperatures by using special alloys: the impact value of austenitic ductile iron at room temperature is 28 ft-lb; at −18 deg C it is 15 ft-lb; at −73 deg C it is 6·5 ft-lb; and at −196 deg C it is still 4 ft-lb. Like ordinary cast iron, ductile iron resists attacks by alkalis, some weak acids, corrosive atmospheres and sea water, the austenitic type being even better than the low alloy grades in this respect; the metal can be nitrided, machined, welded, or silver soldered; further to improve its resistance to corrosion it can be galvanized or electroplated, and to make it still more resistant to abrasion, hard chromium can be deposited on to its surface; it can also be tinned or vitreous enamelled.

Designing for ductile iron has both its easy and difficult aspects. It is difficult in that its various states result in so many useful qualities and combinations of qualities, which are not always readily apparent, that the designer must make himself aware of exactly which type and condition best suits his specific purpose; for this care, he will be well repaid in economy and efficiency; for example, ductile iron freezes at a much lower temperature than steel, thus it is more suitable for the pouring of thin castings; although grey cast iron has the same freezing temperature, it has not the strength and ductility of ductile iron; and malleable iron, although it has good ductility, cannot be used for the pouring of castings in all ranges of thickness, size and weight. The combination of all these characteristics into good design results in economies in both weight and costs: expensive forgings and welded assemblies can be replaced by castings; accurate castings can save machining costs, and the resistance to wear, shock and elevated temperatures allows the replacement of more expensive materials for the same purposes.

In the molten condition, ductile iron has all the good casting qualities of grey cast iron, and when it is solid it has the mechanical properties of steel; thus it can be used for intricate castings of light section that are required to withstand high stresses and rough usage. There is practically no limit to the size or weight of the products, castings having already been produced ranging in weight from 1 oz to 50 tons. Designers should, wherever possible, take advantage of the opportunity to reduce section thickness when changing to ductile iron, as this is made possible by its combination of good mechanical properties and excellent castability. Complete change of design is often desirable and possible, especially when stampings or forgings are replaced by ductile iron castings; solid shapes can be replaced by hollow designs which eliminate unnecessary metal, thus reducing weight and increasing efficiency.

Owing to its castability and good mechanical properties, ductile iron is well suited to the production of pressure castings of many kinds, and its resistance to mildly corrosive conditions makes it especially suitable for parts which are to be used in marine atmospheres. As with all castings, those made of ductile iron should be free from sudden changes of section, and the design should be such that progressive solidification from thinner to thicker sections,

Table 5—SOME TYPICAL GEAR FACTORS FOR S.G. IRON AND OTHER MATERIALS

Materials	*B.S.S.*	*Maximum stress ton/in²*	*B.H.F.*	*Surface stress factor* S_b	*Bending stress factor* S_b
Cast irons:					
Malleable cast iron	300–310	20	140	850	12,000
Grey cast iron, ordinary grade	821	12	165	1,025	6,000
Grey cast iron, high grade	821	22	220	1,450	10,500
S.G. iron, as cast	—	36	240	1,400	19,000
S.G. iron, stress relieved	—	38	240	1,700	19,000
S.G. iron, hardened and tempered	—	55	340	2,500	30,000
Cast steel					
0·4 per cent carbon steel	—	35	152	1,400	19,000
Direct-hardening steels:					
0·4 per cent carbon steel, normalized	En.8	35	152	1,400	19,000
,, ,, ,, ,, ,, surface-hardened	En.8	35	152	2,800	17,000
,, ,, ,, ,, ,, heat-treated	En.8Q	40	179	2,000	24,500
3 per cent Ni steel, heat-treated	En.21R	45	201	2,300	27,000
,, ,, ,, ,, ,, heat-treated	En.23T	55	250	3,000	33,500
3 per cent Ni-Cr steel, surface-hardened	En.23T	55	250	5,100	26,500
2½ per cent Ni-Cr-Mo steel, high-carbon, heat-treated	En.26U	60	269	3,300	36,500
4¼ per cent Ni-Cr-Mo steel, heat-treated	En.30B	100	444	6,000	49,500
Case-hardening steels:					
Low-carbon steel	En.32B	32	140	9,200	40,000
3 per cent Ni steel	En.33	45	201	10,500	47,000
2 per cent Ni-Mo steel	En.35	55	250	11,000	50,000
3 per cent Ni-Cr steel	En.36T	55	250	11,000	50,000
4¼ per cent Ni-Cr-Mo steel	En.39B	85	388	13,000	50,000 +
Cast copper-base alloy:					
Phosphor bronze, sand cast	1400/PB2	12	69	710	7,200
,, ,, chill cast		15	82	880	9,100
,, ,, centrifugally cast		17	90	1,000	10,000

The load-carrying capacity of a pair of gears is determined by the resistance of the tooth surface to failure and by the resistance of the tooth to breakage. In British practice the figures for S_c and S_b are factors inserted in formulae to represent the basic qualities of the gear material, S_c being the permissible surface stress along the line of contact, while S_b is the allowable bending stress. Except for S.G. iron, the data in this Table have been extracted from B.S.S.545:1948, to which reference should be made for further information.

and through them to the feeders, occurs. The pattern-making contraction allowance depends largely on the design of the castings, the composition of the material and its heat-treatment; but, as a general rule, patterns for pearlitic ductile iron castings, as-cast or normalized, should have contraction allowances of about 1 in 90; for annealed, ferritic material from 1 in 120 parts to nil, and for austenitic high alloys about 1 in 55 parts. As with ordinary cast irons, chills can be positioned at the mould surfaces, to cause selective rapid cooling to help to produce sound castings, or to provide zones of considerable hardness for withstanding particularly severe wear conditions during service life.

Stress relief is effected by soaking at about 525 deg C for a few hours; it is a useful treatment for complex castings and has little effect on mechanical properties. Soaking the metal at about 875 deg C for a few hours and cooling slowly in the furnace to 650 deg C, before withdrawing, anneals the material to the fully ferritic condition. If the cooling between 800 and 700 deg C is not slow enough to obtain a fully ferritic structure, this can still be attained by keeping the metal at the lower temperature for up to 8 hours, before continuing the cooling, and then withdrawing; this results in maximum ductility and toughness. Normalizing is effected by soaking at the same temperature and for the same time as annealing, but withdrawing at 800 deg C, then cooling freely in air; this gives high strength, ductility, hardness and shock-resistance. Hardening is effected by soaking the metal, as for annealing or normalizing, then quenching it in warm oil from 850 deg C; this treatment gives good mechanical properties as well as hardness. For tempering after such hardening, the metal is soaked at a temperature of from 200 to 600 deg C, depending on the properties required, as shown in Table 2.

When ordinary ductile iron is austenized by soaking at 850 deg C for up to 2 hours, quenching and then tempering for 1 hour at 350 deg C, the maximum hardness is obtained through any section, resulting in a tensile strength of 65 ton/in^2 and Brinell hardness of 400, but elongation correspondingly only 1 per cent. The hardness obtained by normalizing for 2 hours at 850 deg C is 300 B.H.F., with a tensile strength of 55 tons/in^2. To obtain the hardest surface to the greatest depth the metal should be in the pearlitic condition, and contain from 1·5 to 2 per cent of nickel; in this state a hardness—up to 750 B.H.F.—very much higher than that of white cast iron can be developed to a depth of 0·1 in. Should the metal be ferritic instead of pearlitic, the response is less unless the heating is extended or repeated before quenching. When chills are used in the mould, the graphite in the grey cast iron under the chilled surface of the ductile iron will, on solidification, be spheroidal, with the usual high mechanical properties. In such cases the castings are put into service either as-cast or stress relieved only, because higher temperature treatment will alter the white-iron layers formed in the casting.

In automobile engineering, the annealed grade of ductile iron, to B.S.S. 2789, Type 2A, is used as much as all the other grades put together, although the other grades have their specific uses. This grade is particularly suitable for the production of clutch pedals, flywheels, differential gear casings, back axle housings, gear boxes and housings, carriers, straps and selector forks, pedal levers, wheel hubs and centres, light and heavy duty brake drums, brake shoe segments and cylinders, steering boxes, suspension brackets, trailer hitches, tow bar couplings and hooks, rear spring supports, torque rod mounting brackets, spanners and hand tools, body bolsters for trucks, tyre moulds, fitting machines and tyre irons for tubeless tyres, and motor cycle and scooter frame lugs and brackets.

In the normalized condition, ductile iron finds its best uses in clutch bodies and plates, heavy duty brake drums, brake cylinders, hydraulic cylinders and rams for dump trucks, hydraulic jack bodies, cylinders and rams, press dies and tools, and die shoes. As-cast, to B.S.S. 2789, Type 1, it is employed for hydraulic cylinders and the uses mentioned later for the normalized condition; the annealed, high-ductility grade corresponds to B.S.S. 2789, Type 2B, and is applicable to flywheels, differential gear casings, back axle housings, and gear boxes and housings. As-cast and stress relieved, it conforms to B.S.S. 2789, Type 1—with the addition of the stress relief in the specification—and is used for clutch bodies and plates, heavy duty brake drums, and press dies and tools. Clutch plates, press dies and tools, and trimming dies, are the applications of the quenched and tempered grade, and press dies and tools, as well as trimming dies, are made of a surface-hardened grade.

Booklets on Special Metals

TWO INTERESTING booklets are now available from Henry Wiggin and Co. Ltd., Wiggin Street, Birmingham 16. The first of these, "Wilco-Wiggin Thermometals", gives basic information on the wide range of metals with special thermal properties produced by the company. In addition to a brief description of the properties and applications of each grade, there are six pages of technical data, and information on fabrication and heat treatment.

The second booklet, entitled "Brightray-Coated Exhaust Valves", deals with the application of the Brightray treatment, introduced during the war to improve the service life of the exhaust valves of aircraft engines operating on fuels of high lead content. Originated by the Bristol Aeroplane Co. Ltd., the process can be used either for the treatment of valves during manufacture or for the reclamation of burnt-out valves. The metal used in the Brightray process is an 80-20 nickel-chromium alloy, and the booklet outlines the preparation of valves for its deposition, the applicational method and equipment, and the subsequent machining. Copies of these booklets can be obtained on request from Henry Wiggin and Co. Ltd. at the address given above.

Twist Drill Standardization

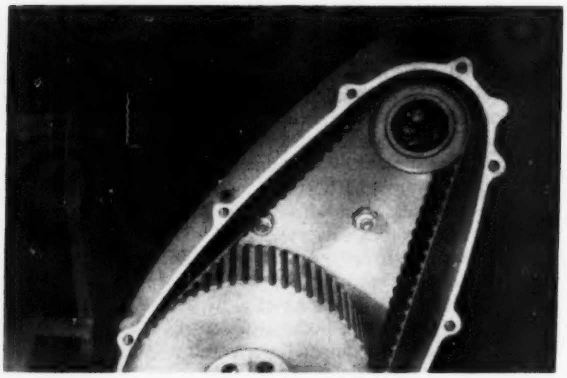
AFTER many years of study, a technical committee of the International Standards Organization has reached agreement on recommended overall and flute lengths for standard types of twist drill. With a view to achieving a more consistently high standard throughout the size range, the relevant I.S.O. Recommendation 301 provides for a more systematic schedule of diameter : flute length ratios than those covered by the various national standards. In the case of the larger sizes of taper-shank drills, the I.S.O. technical committee considered that the flute lengths given in previous standards were excessive for most applications; the shorter drills now specified are more sturdy, with consequently reduced tendency to vibrate during drilling.

A revised British Standard, B.S. 328: Part 1 1959, conforming to I.S.O. Recommendation 301, became effective on 1st January 1960. Other member countries, including Germany, France, Belgium and Italy, are also amending their national standards for twist drills in conformity with the recommendation. Copies of B.S. 328 are obtainable from the British Standards Institution, Sales Branch, 2 Park Street, London, W.1, price 15s per copy.

PowerGrip Drives

Toothed Flexible Belting Now in Use on Automobile Applications

IN THE August 1956 issue of *Automobile Engineer,* a description was given of the U.S. Rubber Company's PowerGrip belt drive system, manufacture of which had then recently been undertaken under licence by The North British Rubber Co. Ltd., of Edinburgh. The outstanding features of the PowerGrip belt are the steel ply base and the Neoprene covering, which embodies teeth on its inner face.



The PowerGrip drive fitted to the Johnson One-20 dumper couples the gearbox to the differential unit. It has a reduction of 2·76 : 1 and transmits a maximum torque of 642 lb-in at 540 r.p.m. of the small pulley, which is flanged to locate the belt. No lubrication is necessary

These teeth mesh with corresponding teeth on the pulleys, to provide a completely positive yet shock-absorbent drive. By virtue of the steel ply, the belt does not elongate in service, so no means of adjustment is needed. Lubrication, too, is unnecessary, though the presence of normal oils has little effect, owing to the resistant nature of the Neoprene. The longitudinal flexibility of the belt enables small pulleys to be used, if desirable, and results in a high mechanical efficiency.

Since their introduction to this country, PowerGrip drives have been adopted for a wide variety of applications, and in the automotive field, these uses include both vehicles and plant. It is known that several car manufacturers are experimenting with the belt for driving the engine camshaft. For this purpose, it has the advantages, over other methods, of compactness of the drive assembly, quietness of running, and the absence of need for lubrication and adjustment. However, the system has yet to go into production for this duty, and the first automotive application in any quantity is to the main drive of a small diesel dumper. This dumper, known as the One-20, is a product of C. H. Johnson (Machinery) Ltd., of Stockport, and is powered by a Lister air-cooled engine, giving 6 b.h.p. at a governed speed of 2,000 r.p.m.

The PowerGrip drive couples the output shaft of the gearbox to the input shaft of the differential unit. A belt of 3 in width and ½ in tooth pitch is employed, and it weighs only 0·090 lb/ft run. There are 21 teeth on the gearbox pulley and 58 teeth on the differential pulley, giving a reduction of 2·76 : 1. When bottom gear is engaged, the belt transmits a torque of 642 lb-in at 540 r.p.m. of the gearbox pulley and, ignoring the low centrifugal loading involved, this results in a belt pull of approximately 385 lb. The drive has a centre-to-centre distance of 12·257 in and is enclosed in a cast aluminium case. As can be seen from the accompanying illustration, the plane through the belt centres is inclined at about 30 deg to the vertical. The driving pulley is flanged on both sides to prevent the belt running off.

Numerous applications of the drives have been made to machine tools, and several manufacturers of quantity-produced cars are using them on transfer lines. For such duties, where a positive drive is needed, the elimination of lubrication and adjustment is again an advantage, and the low noise level is beneficial to the operators. Another valuable asset of this toothed belt arrangement is that the back of the belt can be run on plain pulleys, to modify the line of the drive or to provide a non-positive drive, for example, to a coolant pump. Either a toothed or a plain pulley can be used, on an adjustable mounting, to provide the low initial tension necessary, thereby giving the designer of the machine a greater latitude in respect of centre distances than he would have without a means of adjustment.

D.S.I.R. Technical Digests

RECENT issues of D.S.I.R. Technical Digests include several items of possible interest to the automobile engineer. These are listed below, with their D.S.I.R. reference numbers and the names and addresses of the manufacturers concerned.

1048: Protecting aluminium sheet from handling damage by an easily removable plastics coating. R. A Brand and Co. Ltd., Works Road, Letchworth, Herts.

1056: Gravity conveyor system based on punched steel strip. Dexion Ltd., 65 Maygrove Road, London, N.W.6.

1061: Thermostatically controlled soldering iron. Cardross Engineering Co. Ltd., Woodyard Road, Dumbarton, Scotland.

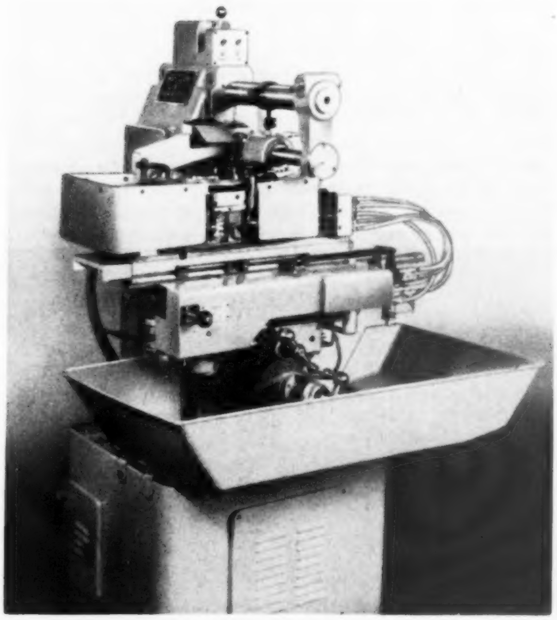
1068: Roller conveyor embodying a safety feature. Manufacturers Equipment Co. Ltd., Sutton Road, Hull, Yorks.

1073: Flexible resistance mat for local heating or temporary small furnaces. Electrothermal Engineering Ltd., 270 Neville Road, London, E.7.

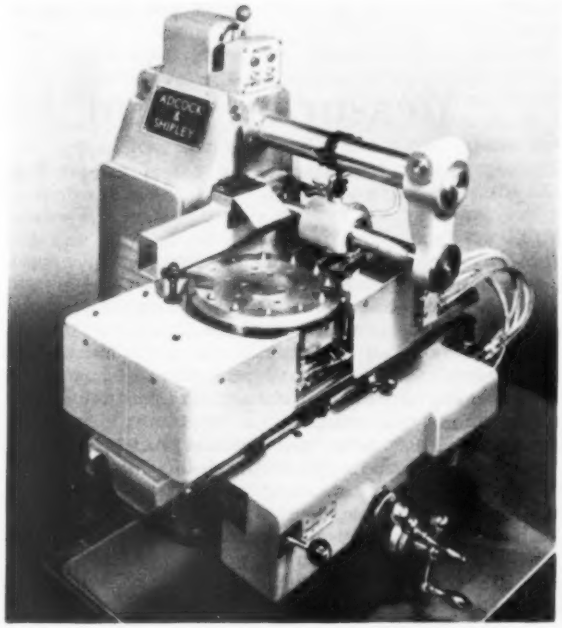
Small Oscilloscope

A SMALL portable oscilloscope has been introduced by Sciaky Electric Welding Machines Ltd. It is known as the Type S.O.1 and is intended mainly for fault finding on production lines, for example, on the control equipment of welding plant. In spite of its small size and the simplicity of its controls, the oscilloscope is claimed to reproduce wave forms comparable in accuracy with those of a laboratory instrument.

The Type S.O.1 unit is designed to operate from a 230-volt, single-phase, 50-cycle a.c. supply, but a tolerance of ± 10 per cent voltage variation is acceptable. Its overall dimensions are $5 \times 3\frac{3}{4} \times 6\frac{1}{4}$ in and its weight is only 5 lb. The address of the manufacturers of this oscilloscope is Falmouth Road, Trading Estate, Slough, Bucks.



Adcock and Shipley, continuous cycling, automatic milling machine



View of work table on Bristol-Erickson air-operated indexing unit

Automatic-Cycle Milling Machine

Adcock and Shipley No. 1 Machine Equipped with Bristol-Erickson Indexer for Continuous Operation

IN its standard form the Adcock and Shipley No. 1, air-hydraulic actuated machine has an automatic table cycle comprising rapid approach, change to feed rate, rapid return, and stop. Motive power for the table is supplied by twin air cylinders mounted on the cross slide. Control of the feed rate during the cutting operation is exercised by a hydraulic cylinder, also mounted on the cross slide.

A new version of this machine is equipped with a specially adapted Erickson indexing table to enable two components to be loaded at one position of the table while two others are being machined at another, diametrically opposite position. The control circuit is modified to give continuous cycling instead of the standard single cycling. Now manufactured by the Bristol Tool and Gauge Co. Ltd., Bristol, the indexing table is also air-actuated.

On the machine illustrated, the workpieces each require two vertical faces to be straddle milled. A gang of four cutters is used and the machine cycles in eight seconds, giving a potential output of fifteen components per minute. Operating continuously, the cycle is as follows:

(1) Table rapid traverse to bring work up to the cutters
(2) Rapid traverse rate changes to feed rate for milling
(3) During the cutting period, an air cylinder operates two plungers to eject two finished components
(4) When milling is completed, the table returns at rapid traverse rate to clear the cutters
(5) At the end of the return stroke, the clamps lift and the table indexes 60 deg.

The operator is required solely to load components into the fixture after ejection of completed work.

Machine table, indexing table, clamping, and the control circuit are all pneumatically operated. Machine and indexing tables are completely interlocked and it is impossible to start the spindle motor unless the air pressure is available for table traverse, indexing, and clamping. Further, it is not possible to start the machine cycling unless the spindle is running.

Following the loading station is a safety station at which misloading of workpieces is detected and then a checking station at which either incorrectly loaded or an oversize part will operate a lever and stop the machine. To ensure maximum rigidity during the cutting operation, both the two components and the table are automatically clamped and are released only during the indexing operation. Automatically ejected parts are delivered into an inclined chute. A spring-loaded wiper arm located between ejection and loading stations removes swarf from the indexing table.

For precise location of the workpieces, the indexing table can additionally be fitted with an outboard plunger. This device is used on the machine illustrated, and the plunger is arranged to operate on a pitch circle of the same diameter as that on which the components are stationed.

Where the period required for loading can be approximated to that of machining, the time saved by this machine can be substantial. Total floor-to-floor time is made up of three component elements only—rapid traverse, milling, and indexing. Although the machine is intended for long, continuous production runs it is readily adaptable to batch production of a range of different components. The table holding the components is secured to the spindle of the Erickson indexer by three screws and located by a dowel, and can be readily interchanged. By holding a number of tables in the tool store, each arranged for a specific component, a changeover of production can be rapidly effected.

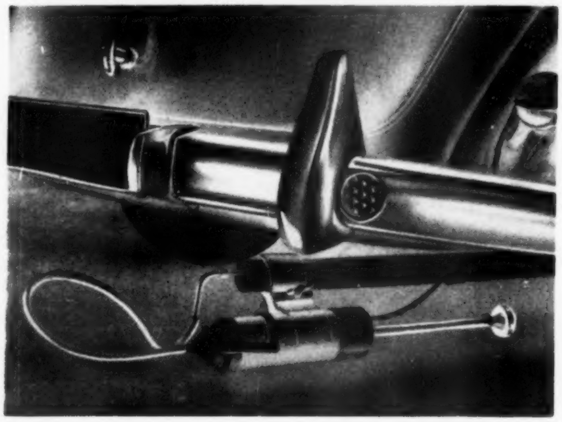
Measurement of Diesel Exhaust Smoke

Robert Bosch Smokemeter: A Development Based on Principles Previously Employed by Saurer

INCREASING public irritation in respect of the black smoke, in some instances ejected with the exhaust gases from diesel powered vehicles, focuses renewed attention on the desirability of accurate means of measuring soot content. While unburned fuel droplets may give rise to blue smoke, only soot is the cause of the black smoke. Owing to the influence of variations in illumination, background and memory, the eye is unreliable for the purpose of measurement, even if it is aided by comparison charts. The effectiveness of a camera for this purpose is limited because only a relatively dark exhaust can be recorded photographically. Carbon monoxide content, although varying with the amount of smoke, and thus providing a medium for measurement, also varies between different combustion systems. Moreover, the procedure for measurement of the carbon monoxide content is complicated and therefore lengthy.

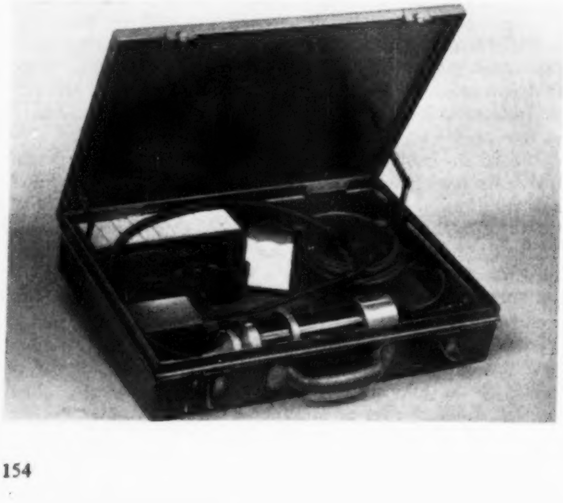
P. H. Schweitzer in the U.S.A. and the Saurer company in Switzerland both developed smokemeters some years ago, while the Swedish Volvo company uses a rig that gives a light absorption reading expressed in percentage of total blackness. Robert Bosch, G.m.b.H., of Stuttgart, have recently further developed the Saurer principle in the design of their smokemeter comprising the two components type-number EFAW 65 and EFAW 68, the former number standing for the accessory case, with an exhaust sampling pump, and the latter for the separate evaluating instrument.

In the Saurer device there are two chambers, used in the manner of an hour-glass: water flows from the upper chamber to the lower, and in doing so aspirates 1,000 cm^3 of exhaust gas through a filter paper. The paper is the filtration medium in a unit from which a probe projects into the exhaust line, and its discoloration is evaluated against a scale. Besides the previously mentioned inaccuracies of visual evaluation, the direct attachment of the filtering unit to the exhaust line introduces problems with regard to condensate formation and unwanted heat, both of which lead to the introduction of inaccuracies in the results. The instrument is hypersensitive, also, to changes in sampling technique and in atmospheric conditions: a few millimetres of water column height above that supported by atmospheric pressure will produce excessive darkening of the filter paper, and the pump will not aspirate properly if the depression in the exhaust line exceeds 100 mm of water column height.



The sampling pump mounted on the tail-pipe on a vehicle. A pneumatic trip lead is taken forward to the driver's position, and the other connection is that between the pump and the probe in the exhaust tail-pipe

Bosch smokemeter accessory case, containing the sampling pump, the probe, and the clamps by means of which the components are attached to the exhaust pipe; also in this box are the pneumatic trip release and packets of filter paper. No other equipment is needed for sampling



With the Bosch smokemeter, these drawbacks are overcome by the use of a powerful suction pump to take the sample; by locating the filter paper remotely from the exhaust line and by photo-electric cell evaluation of the filter paper discoloration. Moreover, its development was directed toward use on the road as well as in the laboratory; accuracy of sampling over a wide pressure range; simplicity of operation, to avoid human errors; quick sampling so that smoke checks could be made with the minimum interference with other traffic, and a price low enough to attract repair stations and authorities concerned with enforcement of standards. The equipment is housed in two steel cases, that containing the sampling pump and accessories measuring $17{\cdot}3 \times 13{\cdot}8 \times 3{\cdot}5$ in and weighing 19·8 lb, while the evaluating instrument measures $11{\cdot}8 \times 7{\cdot}8 \times 5{\cdot}9$ in and weighs 8·8 lb. Only the pump and its allied accessories need be carried on the vehicle, for the contaminated discs can be evaluated later in the workshop or laboratory.

The sampling pump has a 330 cm^3 displacement and the filter unit forms a spring-loaded, hinged cover on the intake end of the barrel, which is remote from the spring-loaded sampling piston plunger. Remote operation is possible by virtue of the use of spring force to take the sample. It is accomplished pneumatically by manual pressure on a rubber air bulb at the end of a 16·5 ft length of hose. Twenty feet of hose should cover most requirements but operation is satisfactory with even greater lengths, if they are needed.

Pneumatic control is effected through the medium of a spring-loaded auxiliary piston, seating against the outer face of the inner cover of the sampling cylinder and operating in its own concentric cylinder, which forms the outer cover of the sampling cylinder. A flexible annular diaphragm clamped to the crown of this auxiliary piston can be inflated by hand pressure on the air bulb, the air entering through a passage through the piston, into which the nozzle is screwed. Normally, spring loading of the auxiliary piston keeps this diaphragm flat between the surfaces, but the pressure exerted by the inflated diaphragm against the inner cover of the main cylinder forces the auxiliary piston away from it.

The sampling piston rod passes through an axial hole in the auxiliary piston, and a knob, for manual operation, is fitted to its outer end. A locking device is incorporated to hold it at the end of its stroke, to set it ready for operation. This device comprises a ring of balls in an annular groove in the axial hole in the auxiliary piston. Normally, these balls are held in their groove by the proximity of the surface of the rod. However, when the rod is fully depressed and the main sampling piston is at the end of its inward stroke, the groove that contains the balls registers with a similar groove round the piston rod; the depth of this latter groove is fractionally less than half the diameter of the balls, which therefore are free to enter it. When the hand pressure on the piston rod is released, a return spring tends to withdraw the sampling piston. The initial motion of the piston carries the balls into a smaller diameter portion of the groove in the auxiliary piston; the depth of this smaller diameter portion is fractionally more than half the diameter of the balls. Thus, the balls are locked in the groove in the rod, and the piston cannot move any further.

When the air bulb is squeezed it inflates the flexible diaphragm on the crown of the auxiliary piston, moving this piston axially, against its spring loading. Because of the restriction the filter unit offers to the inflowing gas, the sampling piston cannot move so quickly as the auxiliary piston; therefore, the movement of the auxiliary piston brings its larger groove into line with the groove in the rod. This permits the balls to escape from their locking position, so that the sampling piston, under the influence of its return spring, can move to the outer end of its cylinder, thus aspirating the sample of gas through the filter paper.

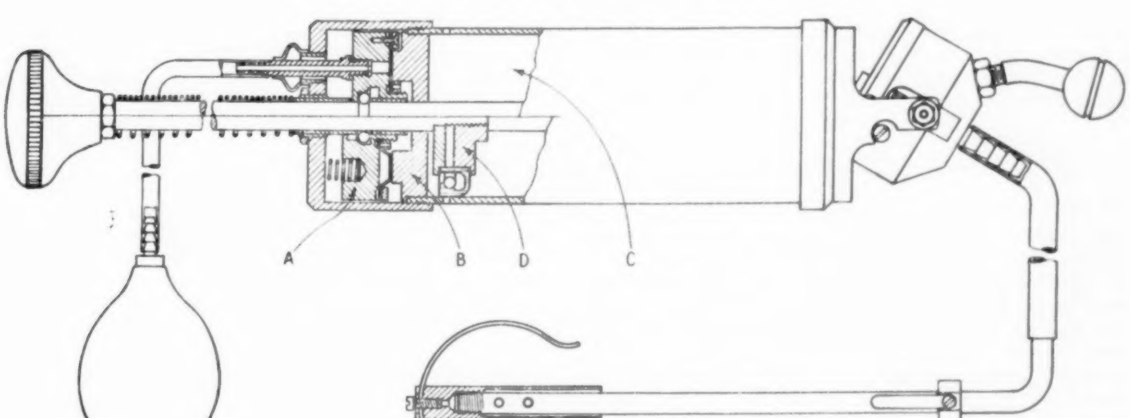
To obviate condensate and heating interference, a 600 mm long pipeline, having an internal diameter of 6 mm, carries the sample from the exhaust pipe probe to the filter disc. The probe is inserted in the end of the exhaust pipe and is located by a U-shape spring, which bears against the exhaust pipe walls. It inhales the gas, through a sleeve, in the opposite direction to the main flow in the exhaust pipe.

For taking a sample on the road, the pump cylinder is clamped to the exhaust pipe by a single wing-nut, the filter paper is inserted in the cap and the sampling piston's plunger is depressed against the spring until the locking device operates. The probe is inserted in the exhaust pipe and the air hose is led to the driver's cab. At the requisite moment, the squeezing of the air bulb releases the plunger and the sample is then available as soon as the vehicle is halted.

The evaluating instrument has a plug-in photoelectric probe containing a light source within an annular photoelectric cell, both being housed in the terminal orifice of the probe. Over this orifice is placed the exposed filter paper, which partially absorbs the light from the source, the amount of absorption varying with the discoloration of the paper. The rest of the light is reflected into the photoelectric cell and forms the basis for measurement of the

Shown in the illustration above is the evaluating instrument, with its photo-electric cell probe and a darkened filter paper. The diagram below is of the sampling pump, the main piston in the upper half of which is locked in the fully depressed position, ready for taking the sample, and that the lower half is in the released position, with the auxiliary piston pushed back against its return springs by the air pressure behind the diaphragm. After the insertion of a new paper disc, the filter unit, shown pivoted ready for this operation, is swung back into line with the sampling pump

A Auxiliary piston; B inner cover of sampling cylinder; C sampling cylinder; D main piston



smoke content of the exhaust. A micro-ammeter measures the current flowing through the cell, zero on the calibrated scale indicating reflection from an unexposed filter paper and 100 representing total blackness. It has not proved necessary to check intermediate calibration, which would, in any case, be almost linear.

A 4·5-volt dry cell located within the casing supplies the evaluator. This cell has a life of several months. In the laboratory, the instrument can be operated from a 12-volt vehicle battery supply, through a voltage cut-down resistance; therefore, no provision has been made for mains operation. This helps to keep the cost down. The cable and the photoelectric probe can be housed within the casing, for ease of transport, and a cover is provided to protect the photoelectric cell when not in use. Response should not fall off for many years, provided that the cell is not exposed to a temperature of more than 80 deg C.

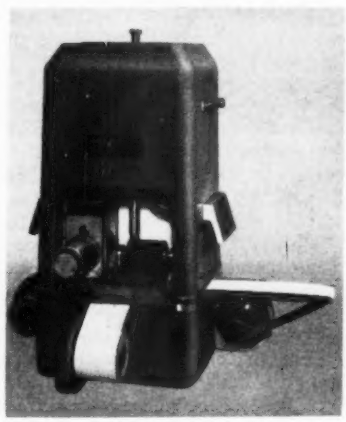
Certain precautions are desirable in laboratory use, where precise results are usually desired. The 600 mm sampling line should be installed so that a portion of the line rises; in repeated sampling this will prevent condensate from reaching the filter paper. If the pressure in the laboratory exhaust line is above that of the atmosphere, a shut-off valve is desirable to prevent soot deposit on the paper before sampling, owing to minute leaks; it is even more convenient to install a venturi between exhaust flanges so that sampling can always occur at sub-atmospheric pressures. Location of the sampling point is not critical, although a sharp bend might introduce unwanted scatter, which is normally held within ± 2 per cent of full-scale deflection of the pointer.

Pitch and Roll Indicator

A Simple Recording Instrument, the Traces from Which Can be Quickly Interpreted

SUBJECTIVE assessment of the ride given by a motor vehicle has several disadvantages, of which the most serious are probably the absence of quantitative criteria, the differences between individual reactions, and the unreliability of memory. Any objective method, involving a system of recording accelerometers, necessitates elaborate experimental equipment, and the subsequent analysis of the records is laborious and lengthy.

Because of these difficulties in ride assessment, Toledo Woodhead Springs Ltd., of Aycliffe, Darlington, has introduced a pitch and roll indicator of an essentially simple nature, the records of which can be interpreted almost immediately after they are made. The apparatus was designed by Plint and Partners, and its simplicity results from a decision to concentrate only on pitching and rolling motions. It was considered that these were among the most important factors in the investigation of riding qualities, and that for certain investigations little would be gained by the added complication of recording yaw or accelerations in the longitudinal, lateral and vertical directions.



In this view of the Toledo Woodhead-Plint pitch and roll meter, the pendulum and the two rolls of recording paper, which feed at right-angles to one another, can be seen clearly

The essence of the indicator is a gyroscopic pendulum, freely suspended with its axis vertical. Any pitching results in a longitudinal movement of the pendulum relative to the vehicle, and rolling produces relative lateral movement. The pendulum is housed in a cast aluminium case, which is firmly mounted in the vehicle. Spark-sensitive recording paper, from two rolls carried on the indicator, is fed horizontally beneath the tip of the pendulum both longitudinally and laterally, one strip overlapping the other. When a record is to be taken, pressure on a button causes sparking to occur between the pendulum tip and the table over which the paper passes. The frequency of the sparks is 30/sec, and they produce a dotted trace on both charts simultaneously. Thus the transverse chart records the amplitude and frequency of any pitching, and the other records the information concerning roll. This ingenious method of simultaneous recording in two directions is the subject of a patent application.

A Varley 24 volt accumulator, of the so-called dry type, supplies the current for the motors of the gyroscope and the paper feeds, and for the sparking equipment. The scale of the records is 4 mm/deg and 20 mm/sec, and the maximum recordable amplitude is 6 deg on either side of each datum line. Measurements of $14\frac{1}{2} \times 14\frac{1}{2} \times 16$ in high are quoted for the Toledo Woodhead-Plint indicator, which weighs 30 lb.

British Standard, S.A.E. and DIN Specifications

BY MEANS of a newly-published set of tables, B.S.3179, Part 1, 1959, ready comparison can now be made between British, American and German standard specifications for the compositions of wrought carbon steels. The tables have been issued by the British Standards Institution to meet the demands of both manufacturers and users of steel. For easy reference, the steels are listed in ascending order of maximum carbon content, thus bringing, as far as possible, similar compositions into close proximity. The British Standard, the S.A.E. and the DIN numbers are given, with the type of product for which the steel is used, its carbon, silicon, manganese, sulphur and phosphorus contents; any additional information is given in a remarks column.

This publication is Part 1 of a series of comparison tables authorized by the Iron and Steel Industry Standards Committee and prepared in collaboration with the British Iron and Steel Federation, in order to meet a demand for a concise form of comparison between British and overseas standards. Part 2 of this standard, dealing with the chemical compositions of alloy steels, will be issued shortly, and an additional table, or tables, based on tensile properties will follow. Copies of this Standard can be obtained from the British Standards Institution, Sales Branch, 2, Park Street, London, W.1, price 6s. Postage will be charged extra.

Brake Design Considerations

An Approach Based on the Concept of a Centre of Pressure for the Reaction Between the Shoe and the Drum

R. M. OLDERSHAW, B.Sc.(Eng.), M.Eng., A.M.I.Mech.E., and A. F. PRESTIDGE, G.I.Mech.E.*

IN the article by J. G. Robinson, which was published in the September 1959 issue of *Automobile Engineer*, application of the work of F. A. S. Acres, and shoe factors are determined for various geometries of the brake shoe, assuming a cosine law for the pressure distribution around the shoe. The following is an alternative approach, and the case of the Lockheed leading shoe (Morris Oxford) is worked out in detail. This approach is based on the concept of a centre of pressure for the reaction between shoe and drum, and on this basis the limitations of the theory, owing to lining lift, are determined. The basic formula for shoe factor, due to J. G. Robinson, is accepted, but the alternative approach that will be outlined shows how the conclusions concerning position and length of lining become modified. This is the only purpose of this report, and no attempt will be made to cover any other considerations that contribute to brake design.

Key to symbols used

For convenience, the symbols used will be the same as those employed by J. G. Robinson, with a few additions. The following list is given for reference:

- OC Centre-line of the brake
- OX Centre-line of the arc of lining represented as MN
- OD Diameter of the construction circle. D lies on OX
- K Centre of pressure
- P Shoe tip load
- R Resultant reaction of drum
- T Tangential force acting at a distance equal to drum radius from the centre O
- F Shoe factor $= T/P$
- ρ Drum radius
- ϕ Angle COX
- α Half angle subtended by lining arc MN at centre O
- β Angle of friction
- μ Coefficient of friction, $\tan\beta$
- l Parameter which determines the diameter of the construction circle, as $OD = l\rho$
- k Parameter that determines the distance apart of the operating and abutment forces, as $EH = 2k\rho$ (Applicable to floating shoe)
- b, c, d Dimensions in a floating shoe, which determine the positions of resultant reaction and abutment reaction relative to brake centre-line OC
- ψ Angle that R makes with OX
- θ Angle XOW, where OW is the direction in which the shoe moves when the brake is applied (Maximum pressure line)
- η Angle with OX, which gives a limit on the position of K

Construction circle

J. G. Robinson refers to the construction circle, and this is the starting point of the work considered in this report. Referring to Fig. 1, the diameter of the circle OD is on the line OX which is the centre-line of the arc of lining. One of the most valuable contributions by F. A. S. Acres was his determination of the diameter of this circle for various conditions of operation.

On the assumption that the brake drum and shoe are rigid whilst the lining is elastic, the pressure distribution around the lining arc follows a cosine law. If $OD = l\rho$, the value of l is given by:

$$l = \frac{2 \sin\alpha}{\alpha + \frac{1}{2}\sin 2\alpha} \quad \text{.......(1)}$$

This condition is generally suitable as a basis for design. F. A. S. Acres represents the centre of the shoe, originally coincident with the centre of the drum, as undergoing a small displacement, to O', so that $v = OO'$. The direction of this displacement is OW, which is the line of maximum

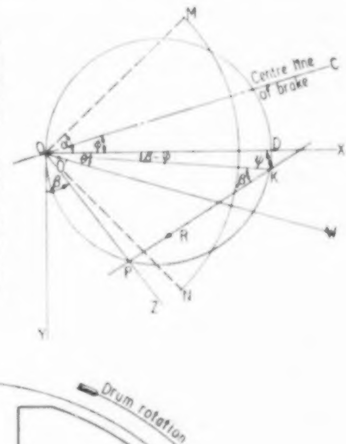
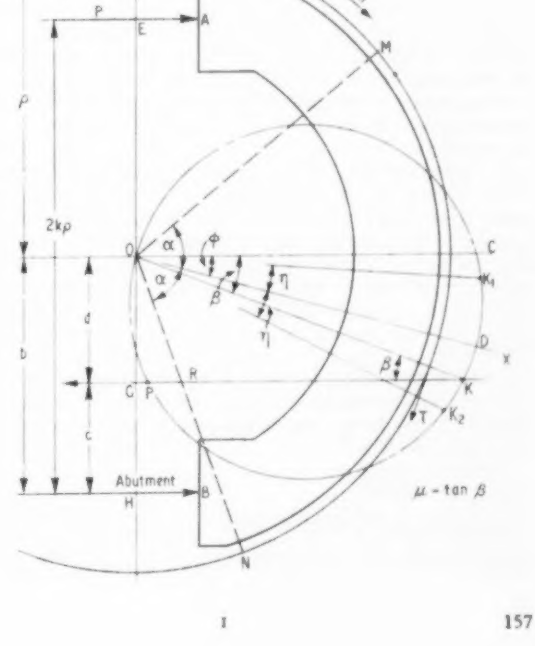


Fig. 1. Right: The resultant reaction R on the arc of the lining GH passes through points K and P on the construction circle

Fig. 2. Below: On a floating shoe, the three forces are parallel. In this illustration, the resultant reaction R, passing through the centre of pressure K, is shown for a leading shoe



*Group Leaders, Research, Lockheed Hydraulic Brake Co. Ltd.

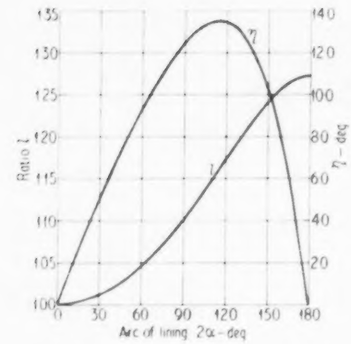


Fig. 3. This graph shows the parameters for lining arcs, and can be used to facilitate computations

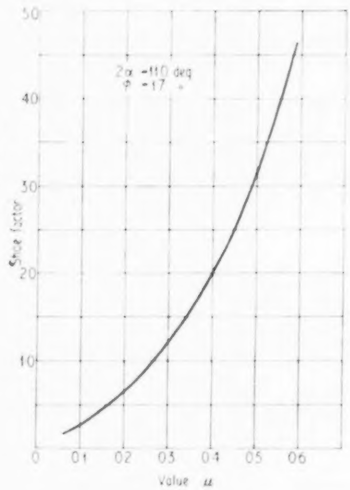


Fig. 4. This curve of F plotted against μ is sometimes known as the stability curve. It holds good only between the limits given by the two equations 7 and 8

pressure. This line is fixed by angle $\theta = \angle XOW$, where

$$\tan\theta = \frac{2\alpha + \sin 2\alpha}{2\alpha - \sin 2\alpha}\tan(\beta \sim \psi) \qquad (2)$$

Properties of the construction circle

The line of action of the resultant force R, of the drum on the shoe, cuts the construction circle at K and P, as indicated in Fig. 1. It was shown by F. A. S Acres that the angle of friction $\beta = \angle YOZ$. Furthermore, the angle β is equal to the angle subtended at any point on the construction circle by chord OP. It is useful to consider point K, then $\beta = \angle OKP$.

K is known as the centre of pressure. The force R always passes through K, and the locus of K is the construction circle. The centre of pressure is thus fixed by the quantities l and β. For convenience in computations, use can be made of the graph shown in Fig. 3, where l is plotted against 2α, thereby giving the diameter of the construction circle.

Shoe factor

Consider now the application of equation (1) to a leading shoe of the floating type. Since the three forces are all parallel to the brake centre-line OC, as shown in Fig. 2, $\psi = \phi$ and $\angle KOC = \beta$.

Then $T \times \rho = R \times d$

and $P \times 2k\rho = R \times c$

$\therefore$ Shoe factor $F = \dfrac{T}{P} = \dfrac{2kd}{c}$

Now, $d = OK \sin\beta = OD\cos(\beta \sim \phi)\sin\beta = l\rho\cos(\beta - \phi)\sin\beta$;

and, $c = b - d = b - l\rho\cos(\beta \sim \phi)\sin\beta$, and so it follows that:

$$F = \frac{2kl\rho\sin\beta\cos(\beta \sim \phi)}{b - l\rho\sin\beta\cos(\beta \sim \phi)}$$

$$= \frac{2kl\rho}{b\,\mathrm{cosec}\,\beta\sec(\beta \sim \phi) - l\rho} \qquad (3)$$

The formula for a trailing shoe is:

$$F = \frac{2kl\rho}{b\,\mathrm{cosec}\,\beta\sec(\beta \sim \phi) + l\rho} \qquad (4)$$

Consider now the particular case, of a leading shoe, when $EO = OH$. Let $b = k\rho$; then a formula identical with that derived by J. G. Robinson is obtained:

$$F = \frac{2l}{\mathrm{cosec}\,\beta\sec(\beta \sim \phi) - \dfrac{l}{k}} \qquad (5)$$

Limitations of shoe analysis

All the preceding results are based on the assumption that the lining is in contact with the drum over the complete length of arc MN. This will not be true if the direction OO' in which the shoe moves, is such that the lining lifts at one end. In this case, the preceding formulae will still apply if only the effective lining arc is considered.

In general, if lining lift is to be avoided,

$\theta + \alpha < 90$ deg

When the limit is reached, say,

$\beta \sim \psi = \eta$

from (2),

$$\tan\eta = \frac{2\alpha - \sin 2\alpha}{2\alpha + \sin 2\alpha}\cot\alpha \qquad (6)$$

This sets limits on the possible location of the centre of pressure on its locus; these are indicated in Fig. 2 as K_1 and K_2. The function for η is plotted in Fig. 3.

Now, $\mu = \tan\beta$

and if $\psi = \phi$ (for a floating shoe)

the position of K_2 gives:

$\mu\,(\max) = \tan(\phi + \eta) \qquad (7)$

and the position of K_1 gives:

$\mu\,(\min) = \tan(\phi - \eta) \qquad (8)$

Equations (7) and (8) set limits on the coefficient of friction between which it can be accepted that the usual shoe factor formulae apply.

Wear considerations

The cosine law for pressure distribution is, to a large extent, backed up by wear patterns observed in practice. Theoretically, maximum material will be removed at the position of maximum pressure, shown as OW in Fig. 1. A designer should, therefore, endeavour to make the position of maximum pressure coincide with the centre of the arc of lining since, otherwise, wastage of material will occur owing to the poor wear distribution. If this condition is to be satisfied, OW coincides with OX, so $\theta = 0$. Then from equation (2) it follows that $\beta = \psi$.

For a floating shoe, $\psi = \phi$; therefore, for good design, make $\phi = \beta$. Then OK coincides with OX, that is, the centre of pressure lies on the centre-line of the arc of the lining. Thus, given the coefficient of friction of the lining, the position of the lining on the shoe is fixed.

Calculation of shoe factor

The Lockheed leading shoe of a Morris Oxford front brake has the following dimensions:

$\rho = 4{\cdot}50$ in

$b = 3{\cdot}38$ in

$k = 0{\cdot}751$

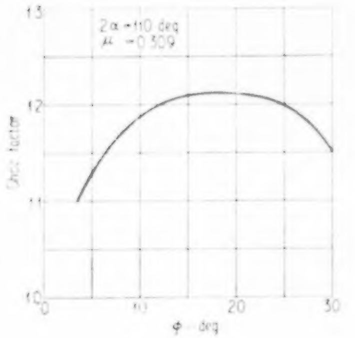
$2\alpha = 110$ deg

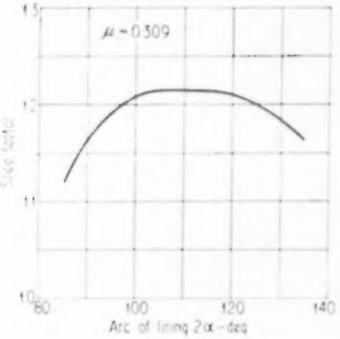
$\phi = 17$ deg

In this case $EO = OH$, and so equation (5) gives the shoe factor for different values of β. The result is plotted, in Fig. 4, as F against μ, sometimes known as a stability curve.

Fig. 6. Left: This graph, showing the effects in change of position of the lining, demonstrates that shoe factor attains its maximum value when $\beta = \phi$

Fig. 7. Right: Curve demonstrating the effect of changing the length of the lining when the trailing end is fixed





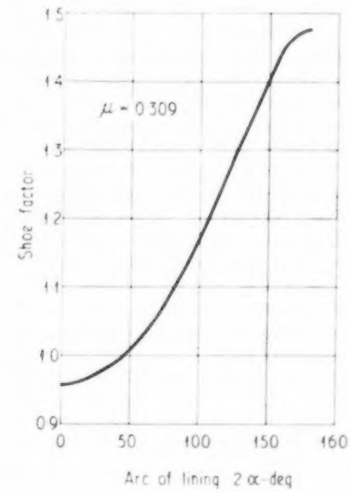


Fig. 5. Curve showing the effect of the change in length of the lining, if the centre-line is in a constant position

The curve is only true between the limits given by (7) and (8). These work out to be:

$\mu\,(\max) = 0{\cdot}589$

$\mu\,(\min) = 0{\cdot}061$

The lining material for which the brake geometry is suited has a coefficient of friction given by $\mu = \tan^{-1}\phi = 0{\cdot}309$. The shoe factor is then 1·25.

Effect of length of lining

Using the above data, the effect of changing the lining arc 2α can be investigated. All other factors are kept constant at the following values:

$\mu = 0{\cdot}309$

$\rho = 4{\cdot}50$ in

$\phi = 17$ deg

$k = 0{\cdot}751$

Again, equation (5) gives the shoe factor, and the result is plotted, in Fig. 5, on a base of lining arc 2α. Since $\theta = 0$, the limits in this case occur at $\alpha = 0$ and $\alpha = 90$ deg. The conclusion is that the greater the lining length the greater is the shoe factor.

Effect of position of lining on shoe

The position of the lining on the shoe was determined by wear considerations. From the point of view of performance, differentiation of equation (3) with respect to ϕ and equating to zero shows that shoe factor is at a maximum when $\beta = \phi$. This is demonstrated by the graph in Fig. 6. The same geometry as above has been applied, with $\mu = 0{\cdot}309$ and $2\alpha = 110$ deg. Then the curve reaches a maximum when $\phi = \beta = 17$ deg. The conclusion is that there is an optimum lining position, given the coefficient of friction; moreover, at this position, performance is a maximum whilst the best wear distribution is obtained.

It may be noted that limits should once again be applied to the range over which the formula used to plot Fig. 6 is valid. From (6) we have $\eta = 13{\cdot}5$ deg; also $\beta = 17$ deg. Then the maximum value of ϕ is $\beta + \eta = 30{\cdot}5$ deg, and the minimum value is $\beta - \eta = 3{\cdot}5$ deg.

Effect of length of lining with trailing end fixed

It is sometimes supposed that a lining should be fixed as near as possible to the trailing end of the shoe, although the preceding arguments demonstrate that this is not the correct approach. Nevertheless, it is of interest to fix the position of the trailing end of the lining arc N, and then investigate the effect of changing the length of arc 2α.

Using the same geometry as before in all other respects, we have $\phi = 72 - \alpha$ deg. Then the result can be calculated, for $\beta = 17$ deg, from equation (5). The graph in Fig. 7 shows the effect on performance. From this it is evident that the maximum shoe factor of 1·25 is obtained when $2\alpha = 110$ deg, this being the original geometry most suitable for $\mu = 0{\cdot}309$.

The limits in this case are obtained as follows. We have, $\beta = \phi \pm \eta$; hence, for $\beta = 17$ deg and $\phi = 72 - \alpha$ deg, the limiting values of α are given by $\alpha - 55$ deg $= \pm\eta$. The solution of this is obtained with the aid of Fig. 3, giving the limits as $2\alpha = 86$ deg, and 135 deg approximately.

Conclusion

On the basis of these considerations, for the Lockheed brake with floating shoes, it is desirable for the centre of pressure to lie on the centre-line of the arc of the lining; only in this way can maximum theoretical performance and best wear distribution be achieved. This conclusion would need some modification when taking into account such things as abutment friction, shoe flexibility and other practical limitations.

The position of the centre of pressure is dependent on the coefficient of friction, which itself is a variable depending on operating conditions. It is necessary for the designer to choose a value for the coefficient of friction which he thinks will be an average during the life of the lining when it is operating under the given conditions; then the lining can be positioned accordingly on the shoe. When calculating stability curves for a particular brake, due account must be taken of the movement of the centre of pressure. If the movement exceeds given limits, the usual design formula is invalid.

References

J. G. ROBINSON: "Brake Design Considerations". *Automobile Engineer*, Sept. 1959.

F. A. STEPNEY ACRES: "Some Problems in the Design of Braking Systems", *Journ. I.A.E.*, 1946-47.

New Plant and Tools

Recent Developments in Production Equipment

ON the "Dormer" point-thinning machine, recently introduced by the Sheffield Twist Drill and Steel Co. Ltd., Summerfield Street, Sheffield 11, only one size of wheel is used for all types of thinning and no special shaping of the wheel is necessary. Symmetrical modification of the drill core is produced automatically. The capacity of the standard machine is two-flute drills from $\frac{1}{16}$ in diameter to 3 in diameter, with straight or spiral flutes, straight or taper shanks, and overall lengths up to 24 in. Provision can be made to accommodate drills of greater length on request.

During grinding, the drill is held stationary and is supported at the point by a conical rest and located by a vee end-stop. A tailstock centre, either male or female as required, supports the end of the shank. The complete holding unit is swivel-mounted on a slide, thus providing universal positioning of the drill. To facilitate the operation, setting positions to produce the manufacturer's recommended core modification are clearly marked for each drill diameter.

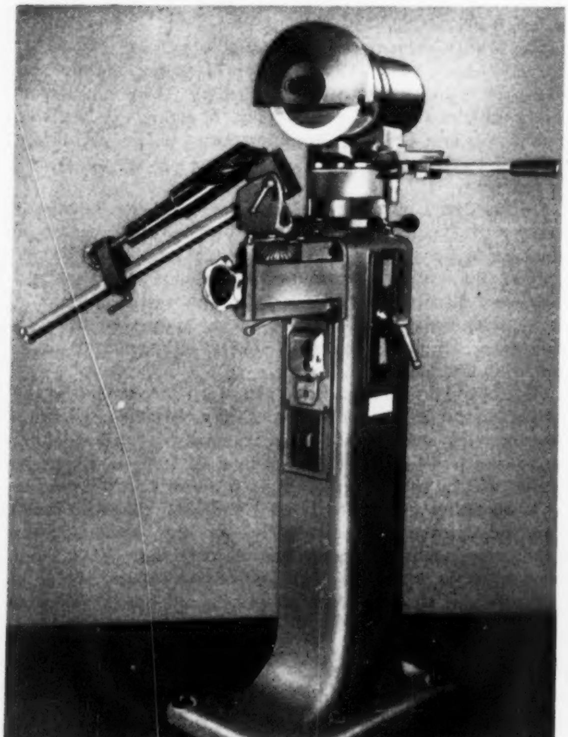
An 8 in diameter $\times \frac{1}{4}$ in grinding wheel is used; driven by a totally enclosed $\frac{1}{2}$ h.p., 3,000 rev/min motor. The wheelhead assembly is mounted on a pivot, enabling it to be manually swung through the drill flute. Feed is automatic; each time the wheel is swung through its arc of movement it is fed forward 0·003 in by a ratchet device. After one side of the drill web has been ground to the desired depth, a stop is locked in position to ensure that the identical depth of thinning is repeated on the other side of the web. Correct shape of thinning is achieved by the setting of the grinding wheel axis in relation to the wheelhead pivot. This adjustment is facilitated by a graduated scale.

Testing springs or checking torque wrenches

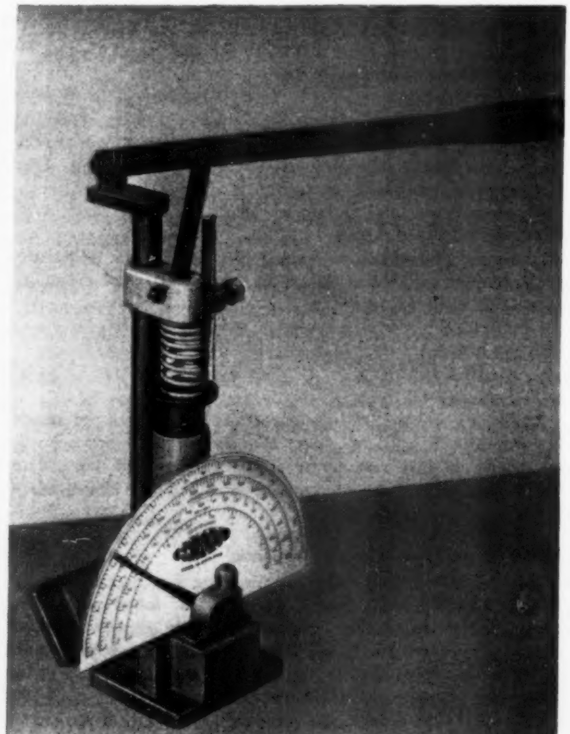
This simple, manually operated tool could meet the needs of testing, development and inspection departments for checking, selecting, or matching helical compression springs —in particular, valve or clutch springs. It has a load capacity of 200 lb and will accept springs up to 4 in long. The base is angled so that the tool can be mounted either vertically, as shown, or inclined at an angle to the horizontal. By detaching the handle assembly from the base and substituting a lever arm attachment it is converted to a tool for checking the setting and calibration of torque wrenches.

A spring to be checked is placed between the upper and lower anvils and compressed to a preset length determined by a stop rod graduated in inches and millimetres. The force applied is transmitted through a master spring to an enclosed rack-and-pinion mechanism which operates the pointer over a substantial aluminium semicircular dial. Separate scales for pounds and kilograms are marked in red.

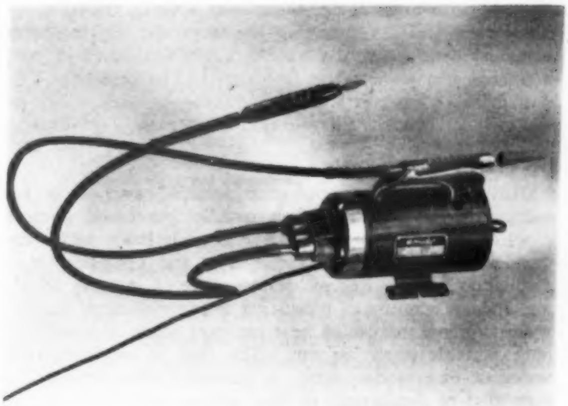
Dormer drill point-thinning machine
(Sheffield Twist Drill and Steel Co. Ltd.)



Compression spring tester, for valve or clutch springs up to 4 in long
(Britool Ltd.)



The attachment to enable torque wrenches to be checked comprises a support bracket on which a lever is pivoted on a needle roller bearing; its free end, furnished with a large-diameter roller, resting on the lower anvil. The wrench is applied to a ½ in square drive socket at the fulcrum point and torque is applied at normal operating speed until the wrench "breaks". Separate scales, marked in black, give direct readings of torque in pounds-inches and kilogram-metres, and reveal any divergence from the setting indicated on the wrench. The manufacturers are Britool Ltd., Bushbury, Wolverhampton.



Morrisflex 300 three-speed machine for three flexible drive shafts
(B. O. Morris Ltd.)

Speed controls for electric motors

For use with motors of from 0·01 h.p. to 10 h.p., Magnedyne controls offer a wide range of speed control with a fine degree of adjustment, a high efficiency, and constant speed, under wide and rapid changes of load conditions, at any one setting. The control equipment is supplied from normal a.c. mains and the d.c. motors are fed from a rectified supply to both the field and armature windings. Speed control is effected by varying the reference voltage applied to the armature, either manually or automatically. Incorporated in the control is a simple electronic amplifier, together with an amplifier of the magnetic saturation type. If the armature voltage drifts from the value selected at any position of the speed control, the electronic amplifier reacts automatically and instantaneously, and thence the magnetic amplifier regulates the energy necessary to maintain the speed of the motor. Equipment is of simple design and robust construction, and is claimed to be trouble free. Only standard low-power valves are used, at well below their normal rating to obviate frequent servicing and replacement. No thyratrons or other gas-filled tubes are employed. Generously rated metal rectifiers are used, to ensure long service without maintenance.

Speed-torque curves are almost flat, and the characteristics, therefore, resemble those of an induction motor combined with the advantages, including ease of control, of a d.c. motor. The design of the control amplifier circuit is such that the maximum current supplied is limited to a given value, normally a little more than that required to give normal working torque. Any mechanical jamming of the motor, therefore, cannot result in any damage either to the motor or to the control equipment. Moreover, "constant-torque" starting is assured. Instantaneous braking is obtained by short-circuiting the armature in the "Stop" position of the control, whilst still feeding the field circuit.

The equipment offers wide possibilities in the field of automation, since the normal potentiometer designed for manual speed control can be replaced by one activated by a suitable programme controller. Alternatively, the reference voltage could be directly controlled by any physical phenomenon that can be translated into the form of an electrical signal. It is also possible to control the speed of the motor to within 0·1-1·0 per cent by means of a tachometer circuit controlled by the motor and feeding back the error signal.

Magnedyne controls are manufactured by S. A. Rochar Electronique, 51 Rue Racine, Montrouge (Seine), France, and marketed by Metrix Instruments Ltd., 54 Victoria Road, Surbiton, Surrey.

Flexible-shaft machine

The new Morrisflex 300 flexible-shaft machine has been developed specifically to provide optimum speeds for steel or carbide rotary cutters, mounted points, and small polishing wheels used for toolroom and similar high-precision work. Manufactured by B. O. Morris Ltd., Briton Road, Coventry, the prime mover is a ¼ h.p. motor complete with a three-speed gearbox giving shaft speeds of 3,000, 9,000, and 15,000 rev/min. Drives to three separate shafts are incorporated, enabling the user to perform varied successive operations without the time-wasting necessity of dismounting and mounting different tools. Each flexible shaft is

Magnedyne electronic speed-control equipment for electric motors of up to 10 h.p.
(Metrix Instruments Ltd.)

complete with a handpiece fitted with a 6 mm collet.

To meet varied or varying requirements, the motor is furnished with a carrying handle, a foot for bench or wall mounting and an eye-bolt for overhead suspension. The manufacturer can supply an exceptionally wide range of tools and equipment for the skilled operator.

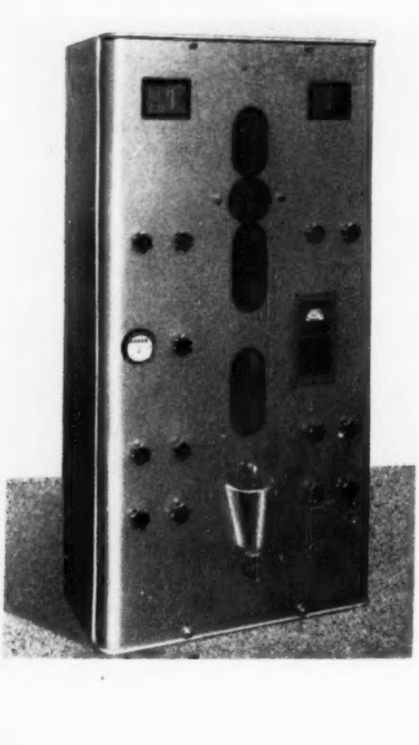
De-ionized water supply

Distilled water for topping-up vehicle batteries, or for making battery electrolyte, is usually purchased from a supplier since the rate of consumption is likely to be less than would warrant the installation and operation of a distillation plant. Recent development of systems of ion exchange as a means of producing the equivalent of distilled water without the use of heat has now made it possible to have suitable water on tap. The cost of producing this water of exceptional purity is generally considerably less than that of distillation; in fact, gallons per penny instead of pence per gallon.

The process consists of passing the water through a vessel containing a bed composed of two types of synthetic resins that remove all the bases and radicals from the water. Water obtained by this process, in particular by means of the Paterson Solobed system, is of a quality much higher than it is possible to obtain from single distillation. The only satisfactory method of ascertaining its purity is by measuring the resistance of the water to electrical current.

For the regeneration of the resins both acid and alkali are used and, therefore, the construction of the equipment must be both acid- and alkali-resistant. In the smallest Solobed unit manufactured by the Paterson Engineering Co. Ltd., 129 Kingsway, London, W.C.2, which gives an output of 10-15 gallons per hour, the pipework is of plastics, valves are made of ebonite and other parts from transparent plastics materials so that no metal is in contact with the water. This is an essential feature since the water produced is of so high a purity that the most minute traces of contaminants have a marked effect on its conductivity.

Solobed water de-ionization unit
(Paterson Engineering Co. Ltd.)



Major high-speed, tube-sawing machine
(Addison Tool Co. Ltd.)

Tube-sawing machine

The concept of using HSS saws for cutting tube stock at production rates equivalent to those for abrasive wheels, is exemplified by this Major tube-sawing machine. It is claimed to give a cleaner cut, to obviate the need for subsequent deburring operations, and to yield a reduction in production costs. The capacity of the machine illustrated, having an 8 in saw blade, is tubes up to 2 in outside diameter.

A cast wheelhead casing houses the 1·5 h.p. driving motor and incorporates the oil bath in which the clutch and wheel spindle are immersed. The complete wheelhead is mounted on vee slides to enable it to be secured in the cutting position best suited to the diameter of the work, and adjustment is also provided for taking cuts at angles up to 45 deg. Equipment includes a lever-operated tube vice and a spring-loaded device to support the cut-off tube on the right-hand side of the saw blade and thus prevent the formation of a fall-off burr. Coolant from a tank in the pedestal is circulated by a pump driven by an independent motor. On actual production runs, up to 2,000 cuts can be made before the saw blade requires resharpening.

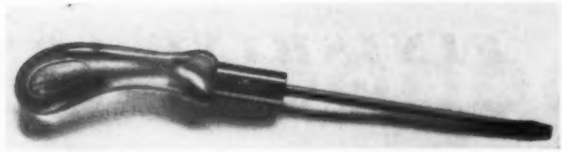
Manufactured by the Italian firm of Pedrazzoli, in Basano, the sole distributing agents are the Addison Tool Co. Ltd., 28 Marshalsea Road, London, S.E.1.

Transistorized sound-level meter

In view of the attention currently being directed to the reduction of noise, the introduction of the Type 1400E, fully transistorized, sound-level meter by Dawe Instruments Ltd., 99 Uxbridge Road, London, W.5, is timely. Claimed to be the first unit in the world of this type to be made commercially available, it is completely self-contained, can be operated readily with one hand, and is conveniently portable. Powered by built-in dry batteries, its overall dimensions are $8\frac{1}{4}$ in $\times$ $5\frac{3}{4}$ in $\times$ $3\frac{3}{4}$ in, and its weight is less than 4 lb.

When the instrument is not in use the microphone is folded down into a recess in the top of the case. On raising the microphone to the working position, the meter is automatically switched on and a reading can be taken imme-

Dawe Type 1400E transistorized sound-level meter. Range 24 to 140 dB
(Dawe Instruments Ltd.)



Steadfast ratchet screwdriver with pistol-grip handle
(J. Stead and Co. Ltd.)

diately, since transistors require no warming-up period. One set of batteries provides an operating period of 60 hours.

Apart from a microphone and high-gain amplifier, the instrument incorporates a variable attenuator to give a direct reading of sound level over the range from 24 to 140 dB. The standard meter, therefore, covers the range for virtually every industrial application, except possibly for measurements in the near vicinity of turbojet engines, for which a special microphone must be substituted. A further feature is the inclusion of weighting networks to simulate the frequency response of the human ear. The meter reading gives, therefore, a close measure of the equivalent loudness of all sounds without the possibility of error arising through age, prejudice, or fatigue, to which all human observers are susceptible to some extent.

Typical applications of this meter in the transport industry include the checking of permissible noise levels of motor vehicles, the measurement of noise levels in passenger compartments to ensure maximum comfort, the inspection of engines, gearboxes, and transmissions for operating smoothness, and the study of road surfaces and tyres. All conventional uses in workshops, test rooms, and offices, of course, are well within its operating range. The Type 1400E meter meets the proposed International Electrochemical Commission (I.E.C.) specification for sound level meters.

Ratchet screwdriver

Even the ubiquitous screwdriver is receiving attention with the aim to make it easier and more reliable in operation, either for general or specific usage. An example of interest is the new ratchet driver introduced by J. Stead and Co. Ltd., Cricket Inn Road, Sheffield. The blade of standard pattern, with a flared point, is made from chrome-vanadium steel and is chromium plated to resist corrosion; the ratchet mechanism is robust and positive.

Novelty occurs in the shape of the unbreakable handle moulded in a translucent, amber coloured, plastics material and providing insulation that effectively withstands a current of 5,000 volts. Instead of following the familiar cabinet, circular, or polygonal patterns, it is given a slightly angled, pistol-grip shape. This makes it unusually convenient to handle, and easier to drive home or loosen tight screws. Particularly, it facilitates manipulation at arm's length or in closely confined applications as are frequently encountered beneath a chassis or in the interior of a body.

Multiple barrelling machine

The interest of Rolls-Royce Ltd. in barrelling technique has earlier been referred to in these columns.* A new range of multiple barrelling machines for small, batched work has been developed by Rolls-Royce Ltd. and is now being marketed by Roto-Finish Ltd., Mark Road, Hemel Hempstead, Hertfordshire. Three models have been standardized, with basic units of six, nine, and 12 barrels or drums respectively. In each case, however, a further drum unit which doubles the capacity can be added if desired. Drums are of a plastics material and have a capacity of $\frac{1}{4}$ ft^3.

Each machine has a number of vibrating screens, chip bins, a sink for washing-out, and a drawer for housing the special tabletted compounds for use in processing. The precise dispensing of compounds in tablet form is another Rolls-Royce development, and was briefly described in the quoted article.

Individual control of processing time for each drum is by means of a separate time switch. At the end of the time cycle, the cradle holding the drum is tilted forward, indicating that processing is completed. By selection on a dial wheel on the front of the machine, the grinding chips are returned after processing to the appropriate chip-size container for re-use. The machines are wheel-mounted and readily mobile so that they can be moved to any shop location.

*Precision Barrelling "Automobile Engineer", December 1958.

Multi-Matic self-contained, 24-drum, multiple barrelling machine
(Roto-Finish Ltd.)

FINISHING CONNECTING RODS

American Surface-Grinding Practice is Shown in Specially Equipped Mattison Machines

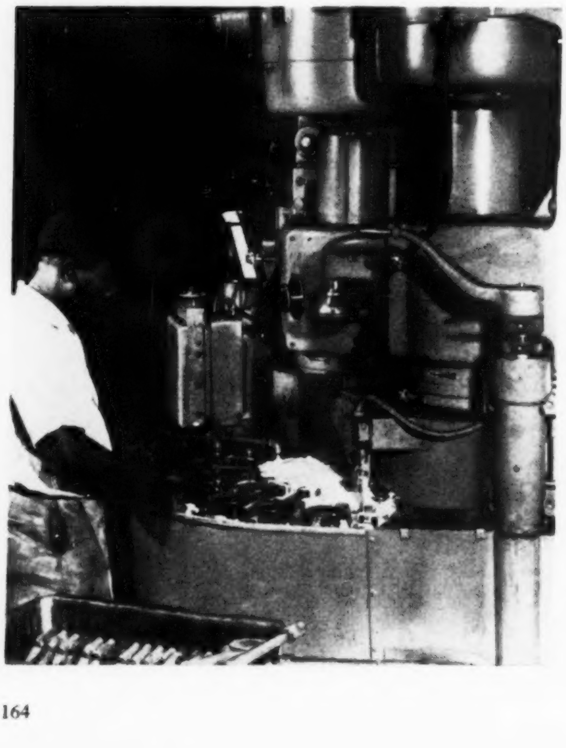
SURFACE grinding operations on connecting rods include the finishing of the big-end joint surfaces, where they are engaged by the big-end cap, and the finishing of both side faces of the rod after the cap has been assembled to it. For such operations, the American view is that no method or machine has yet been found to surpass the modern vertical, multi-spindle, multi-work station, automatic surface grinder, either as regards the rate of production or in reference to the quality and consistency of the work. The high performance of machines of this type is due in no small part to the excellent automatic clamping facilities they incorporate. Where properly carried out, the design and application of an automatic clamp depends mostly on careful analysis of the operation to be performed. Clamps developed for use on some of these machines, engaged in processing connecting rods, are as simple as is possible consistent with the work they have to do.

The fixtures in which connecting rods are clamped are arranged in ring formation on an annular work table. This table rotates about a massive, vertical, central column. Vertical spindles of the grinder, together with their motors, are arranged around this column, so as to leave an open space for loading and unloading the fixtures at the front of the machine. Whether a machine is tooled for grinding the joint surfaces for the connecting-rod cap, or for grinding side surfaces of the rod, either roughly or to finish size, clamping fixtures are designed in such manner that surfaces to be finished are held in a horizontal plane.

Stationary cams are mounted around the inside of the heavy rail which encircles this rotating, annular table, to form a discontinuous cam track. Usually, each clamping fixture is operated by a pivoted lever furnished at the end with a roller-type cam follower. Immediately before the workpiece that the fixture contains encounters the first grinding wheel, the cam forces the follower radially inwards against the resistance of a compression spring. Inward movement of the lever produced in this manner is used, in one way or another, to provide the necessary clamping force.

Automatic clamp devices for the grinding of rod joint faces

Mattison two-spindle machine grinding connecting rod side faces



After making a complete circuit of the central column, a given clamping fixture returns to the unloading station. At this point, the cam follower emerges into the open and runs out of contact with the cam. Thereupon, the compression spring reacts, opening the fixture for the ready unloading of the part. The connecting rod or other workpiece in that particular fixture is then removed, and replaced with another component to be ground.

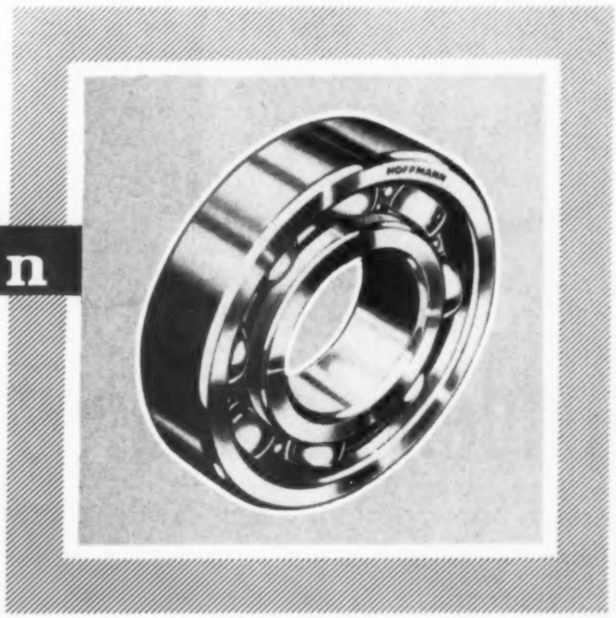
In some instances, Mattison rotary grinders of this type are equipped with only two vertical spindles, and two separate surface-grinding operations are performed on each side of the connecting rods with their caps attached. In some applications two such machines are employed. The first is used to rough-grind both sides of the drop-forged connecting rods, before the caps are severed.

Finish-grinding on the sides of the rods is delayed until after all drilling, broaching, cap parting, bolt insertion, and re-assembly operations have been completed. Then, the rods are routed to the second two-spindle machine, where both sides are finish-ground to size. By handling the work in two stages in this manner, it becomes practical to use a segmental wheel for the rough-grinding operations and a standard wheel for finishing. The normal wheel, of course, makes it possible to obtain a finer surface finish and to hold dimensions to a somewhat closer tolerance.

The two-spindle machines illustrated, and the special clamping fixtures in each case, were manufactured by the Mattison Machine Works, Rockford, Illinois, U.S.A.

instant at the **response** touch of a button

Hoffmann



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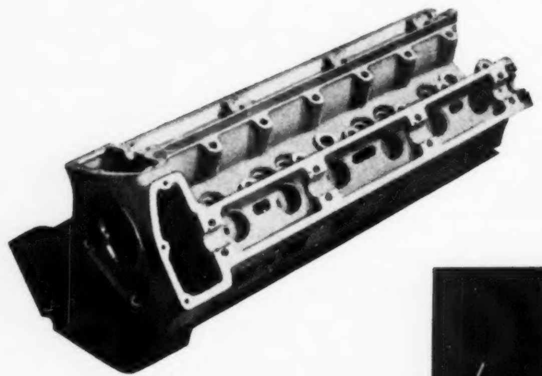

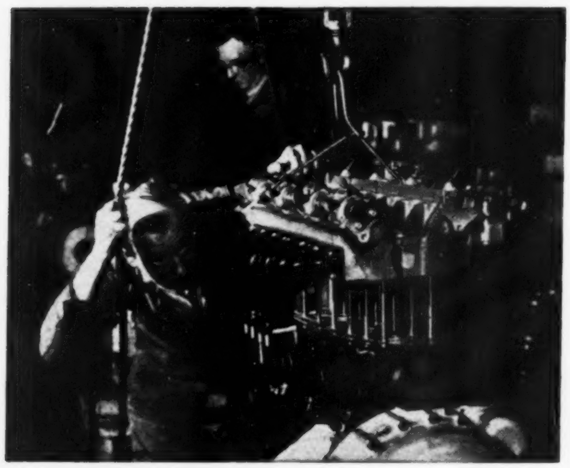

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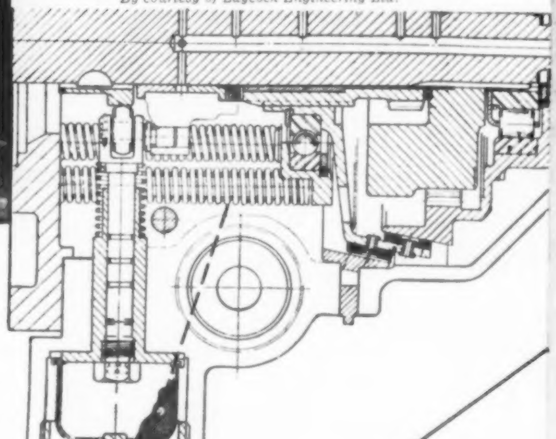

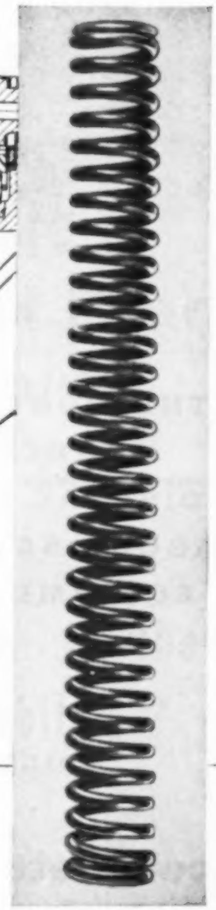

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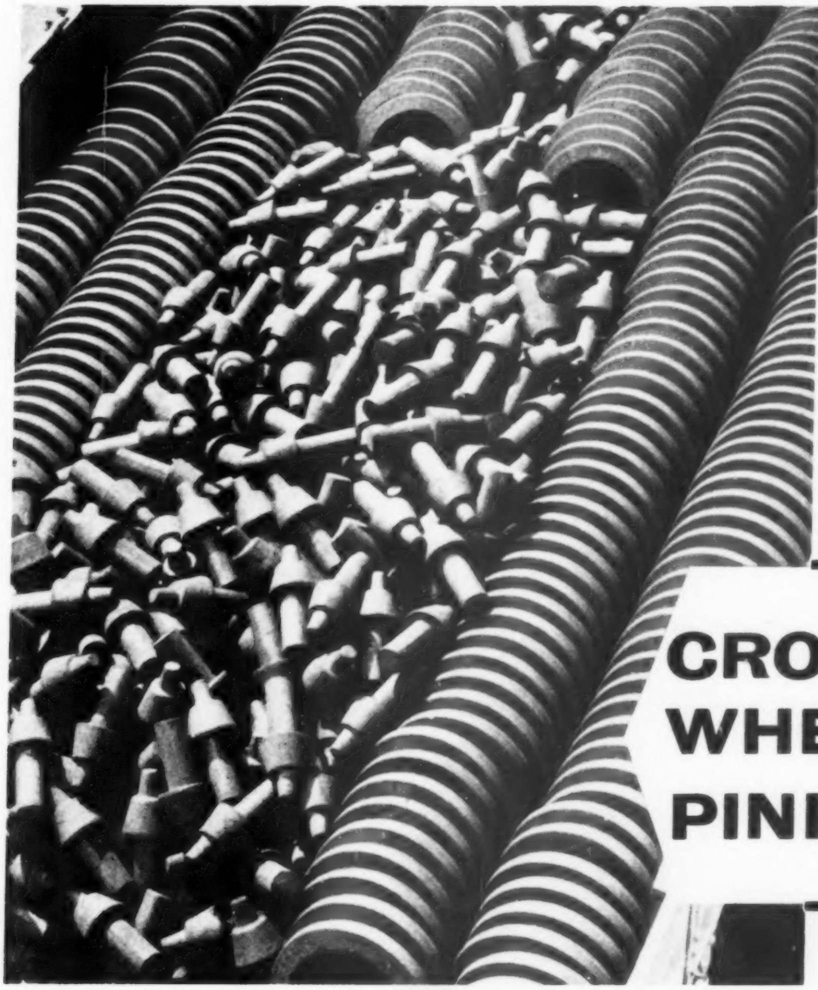

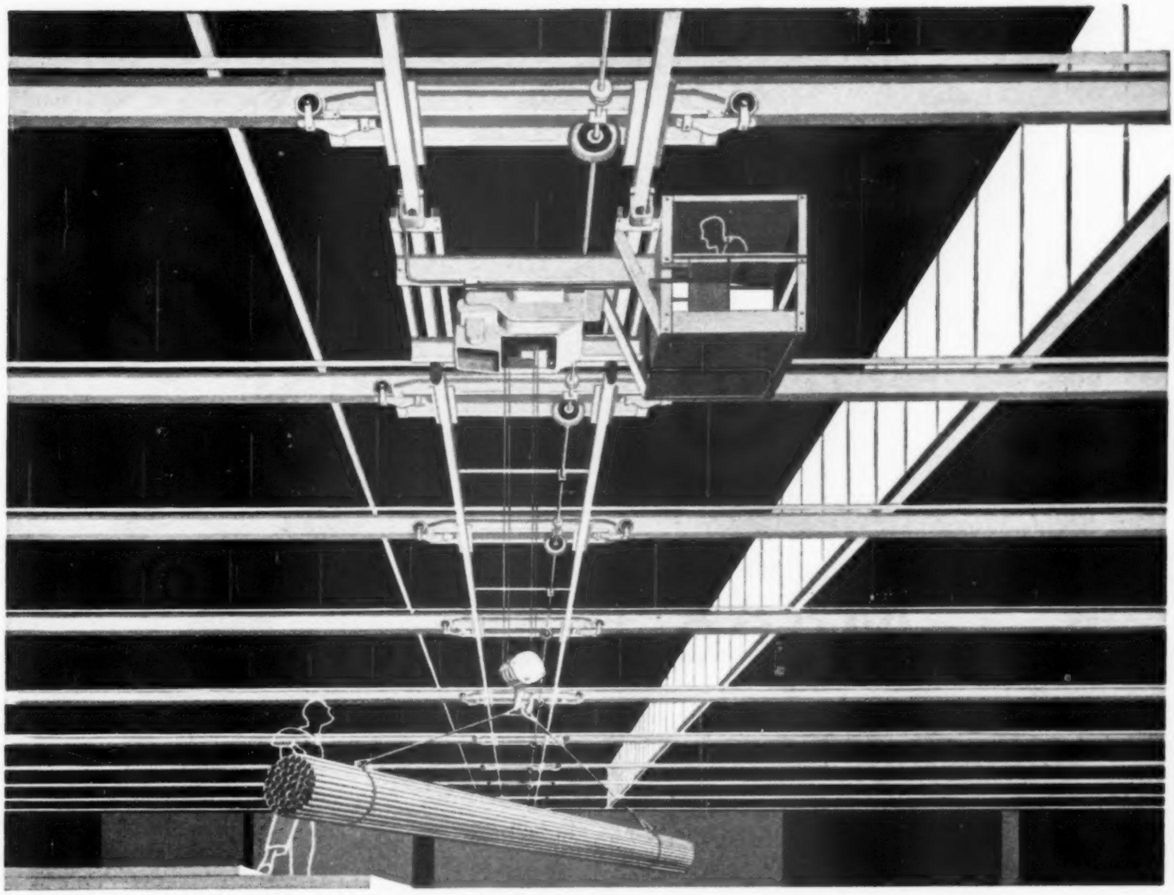

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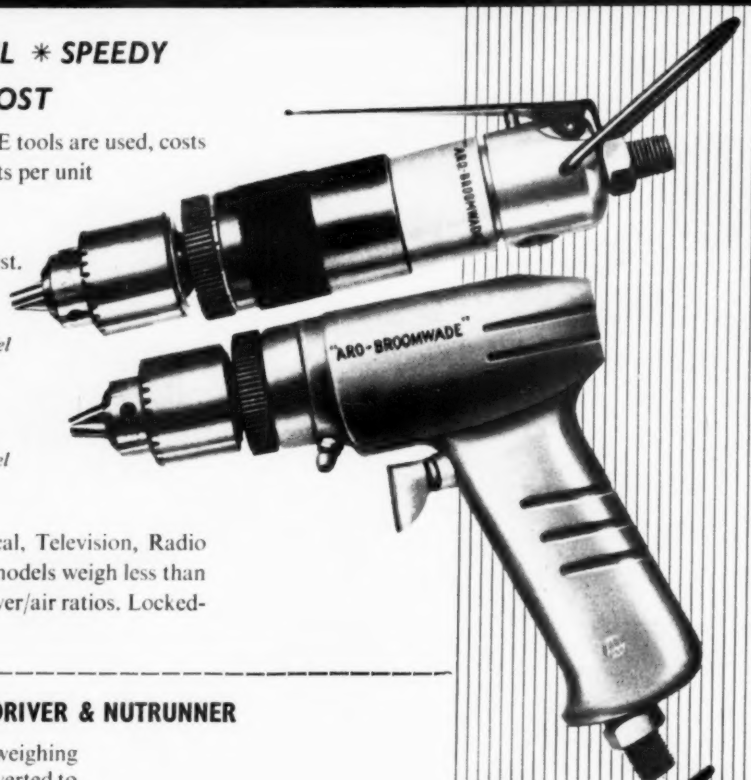

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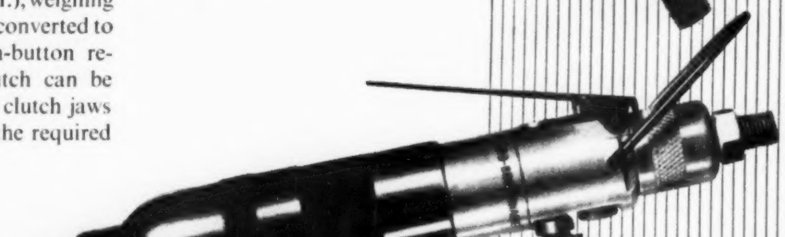

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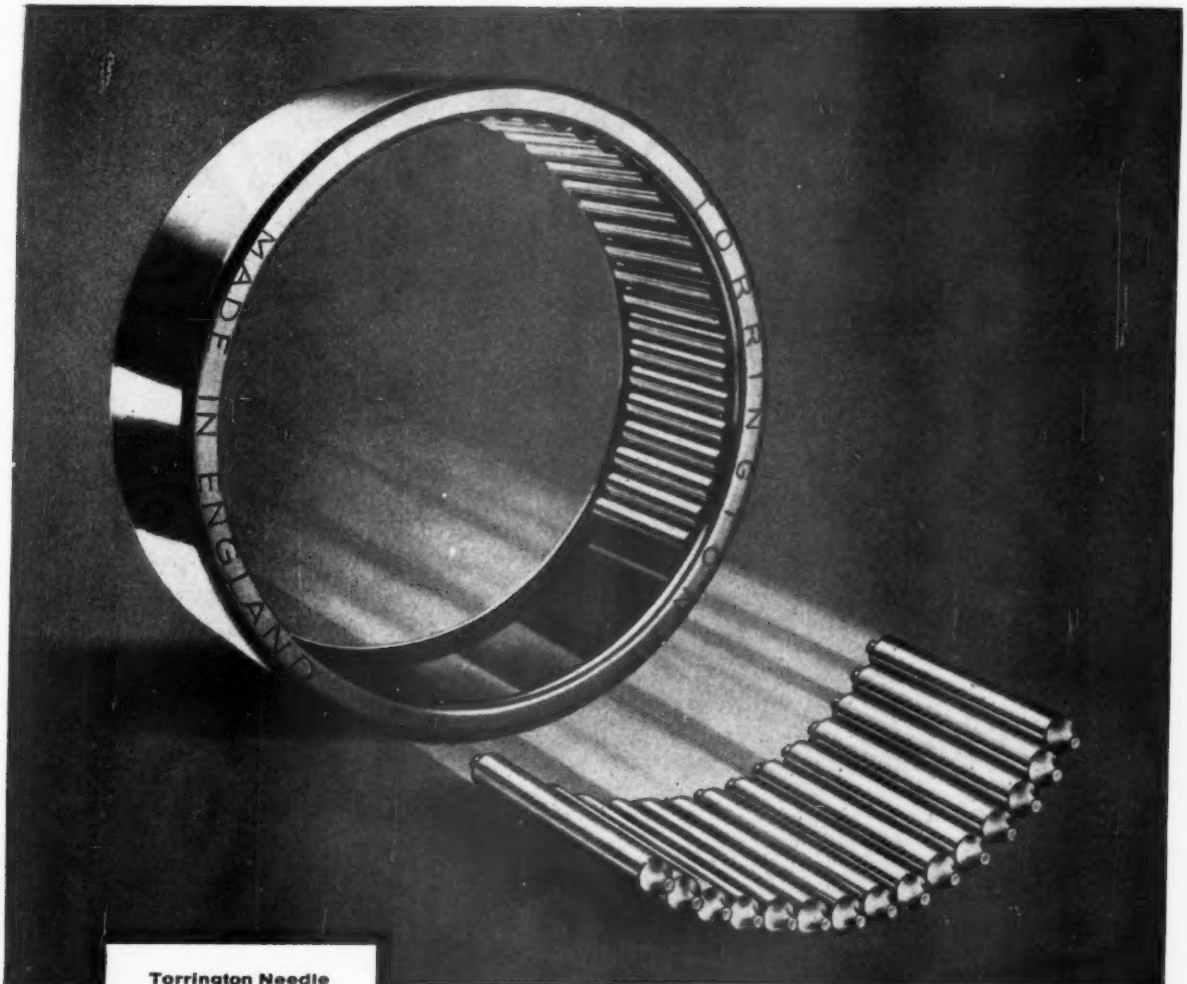

Torrington Needle Bearings feature:

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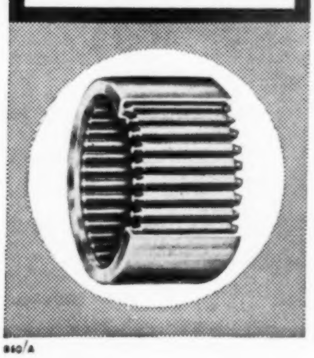

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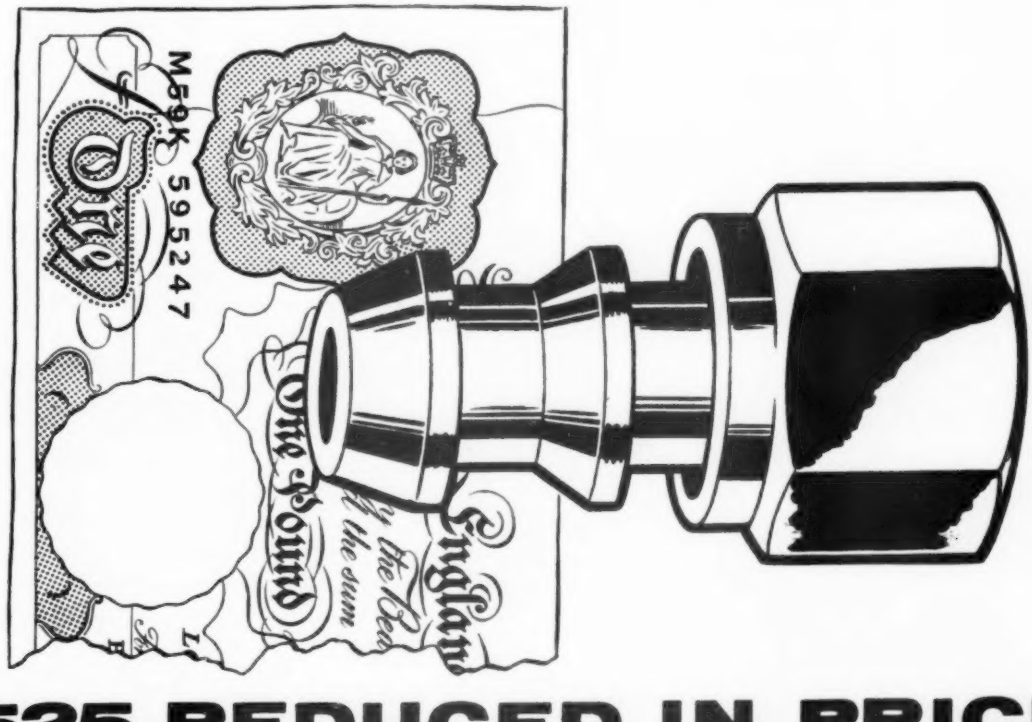

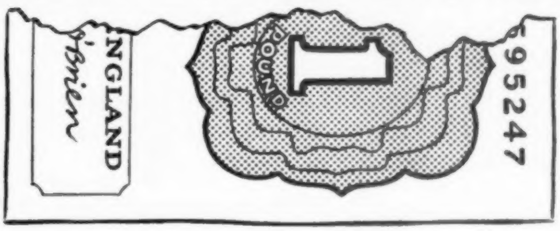

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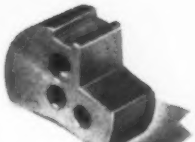

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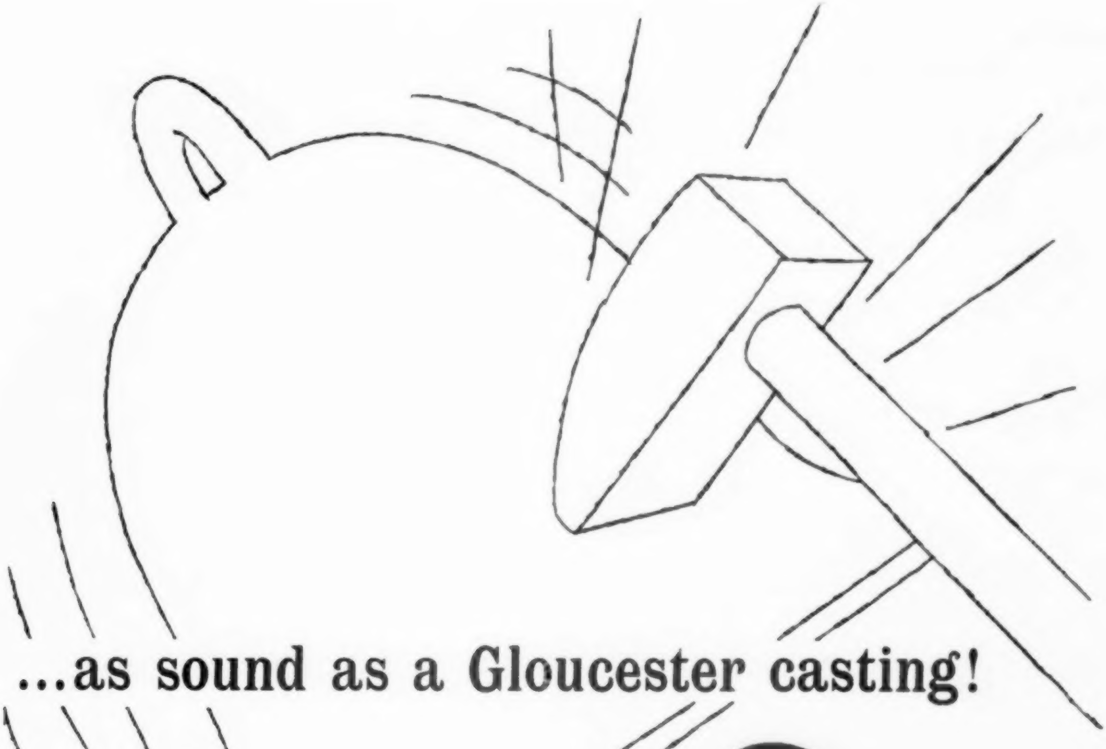

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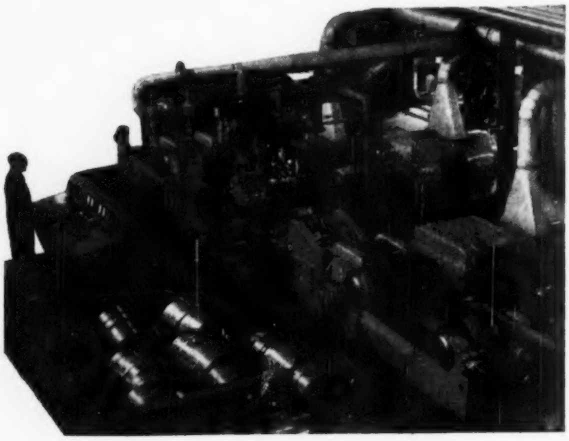

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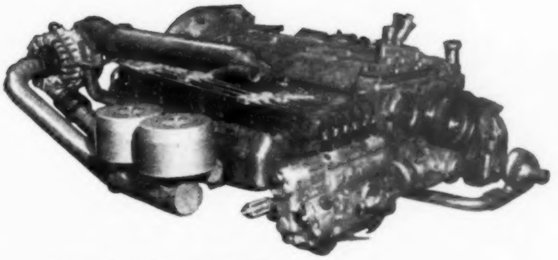

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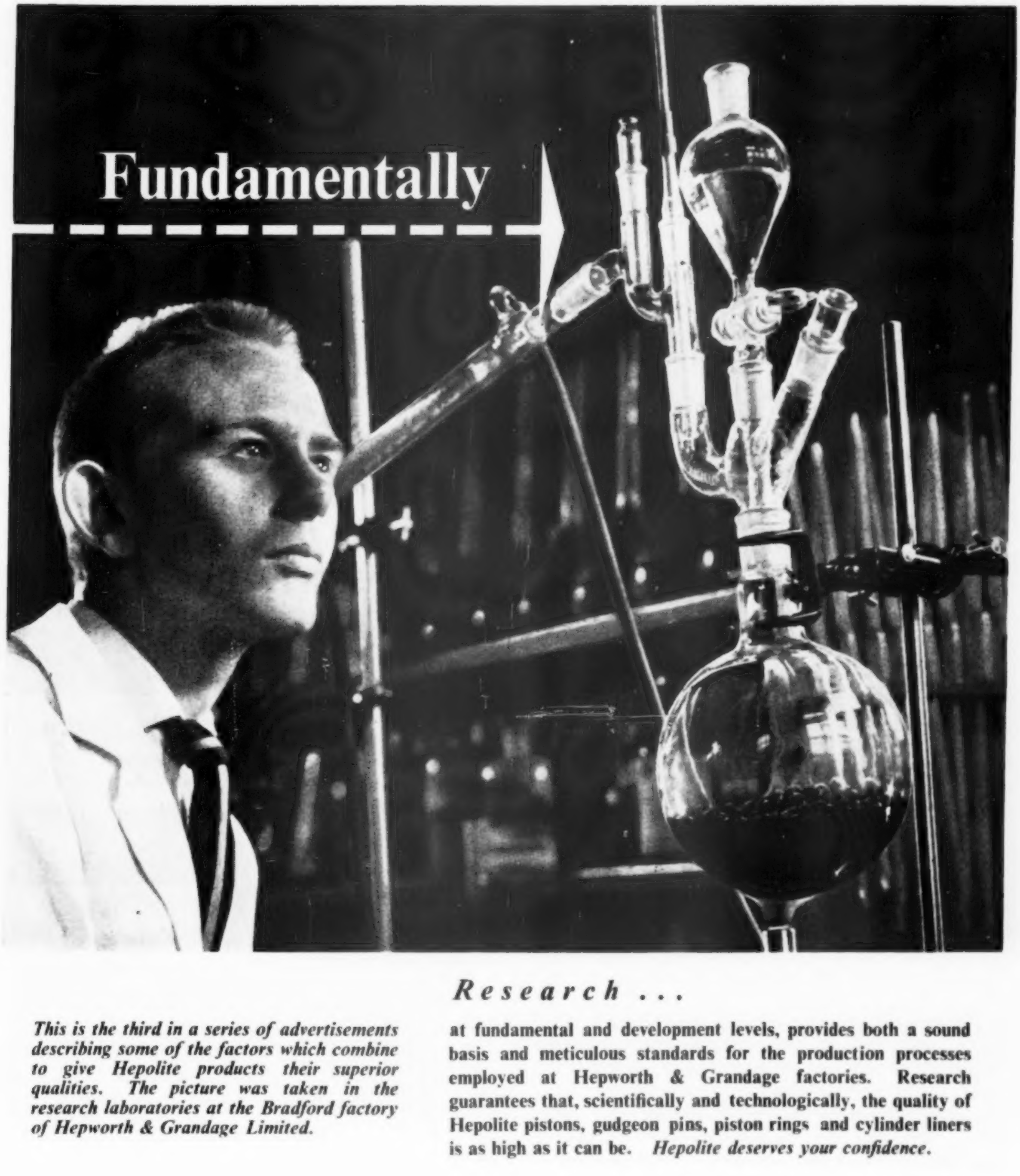

Research . . .

This is the third in a series of advertisements describing some of the factors which combine to give Hepolite products their superior qualities. The picture was taken in the research laboratories at the Bradford factory of Hepworth & Grandage Limited.

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Hepolite engine components are fitted as standard equipment in most British cars, and the range of these items available from Hepworth & Grandage Ltd., is the most comprehensive in the world. ***Please write for details to Dept D.***

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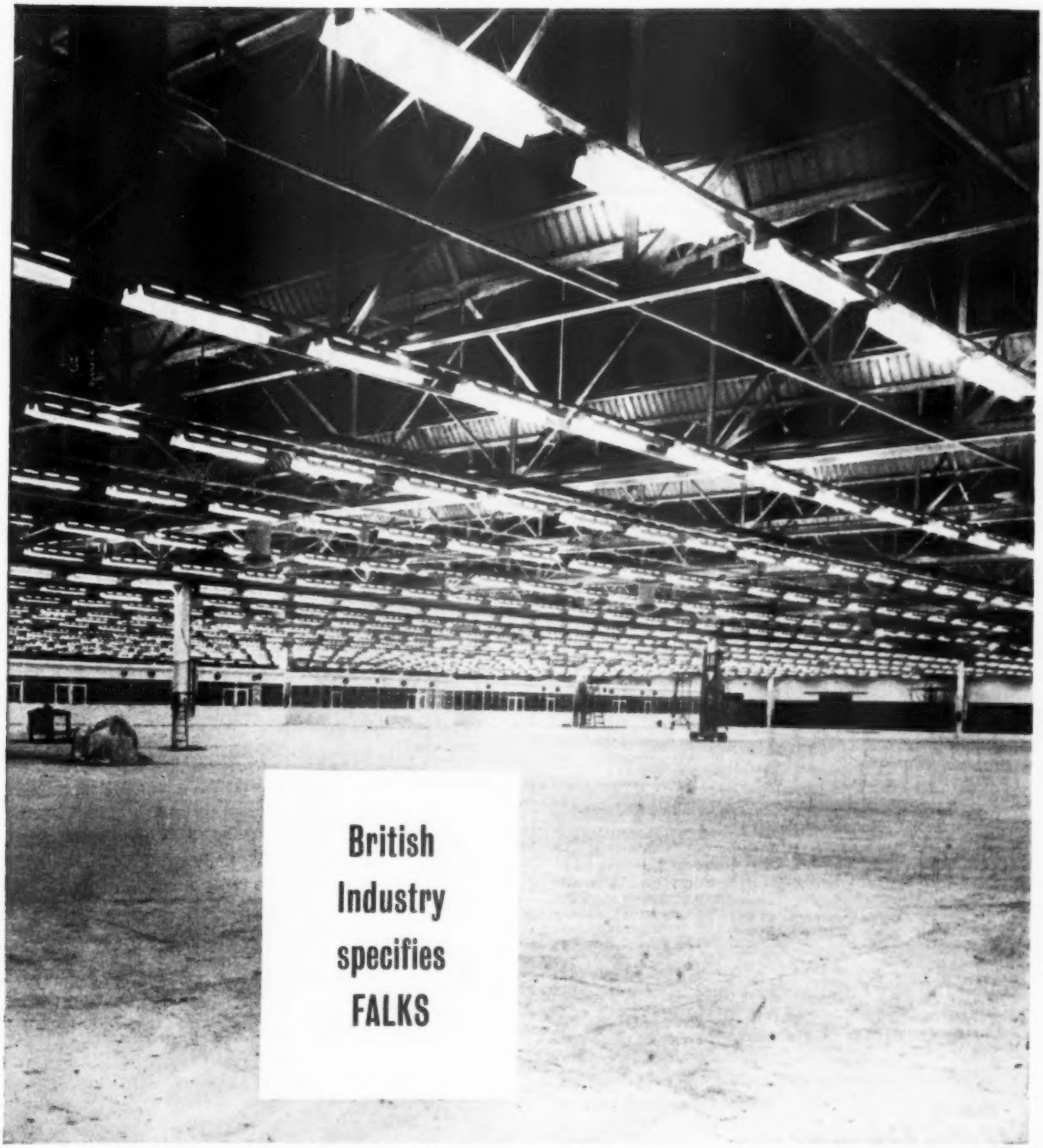

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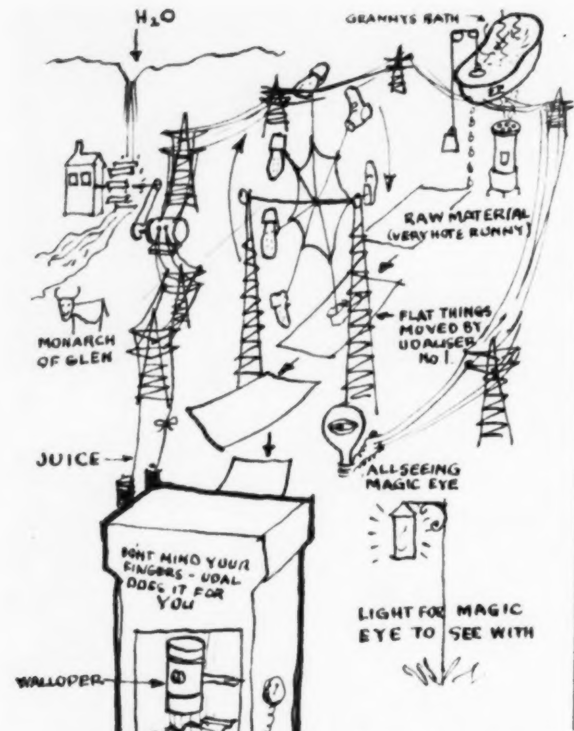

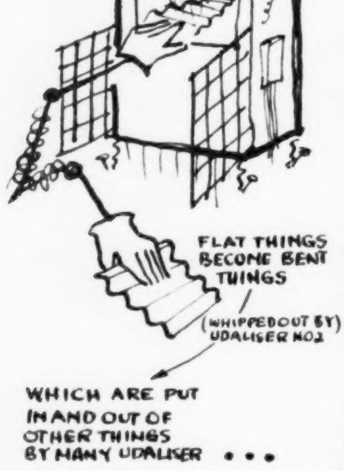

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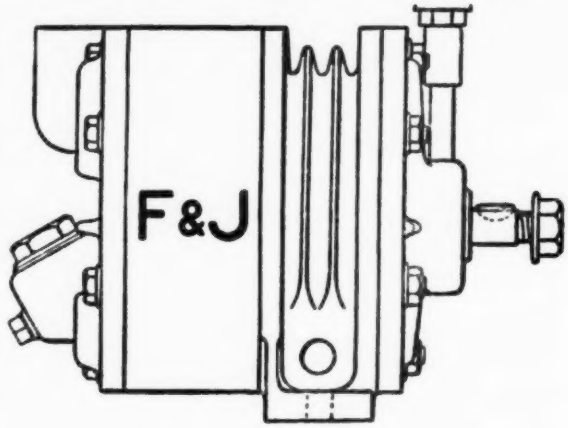

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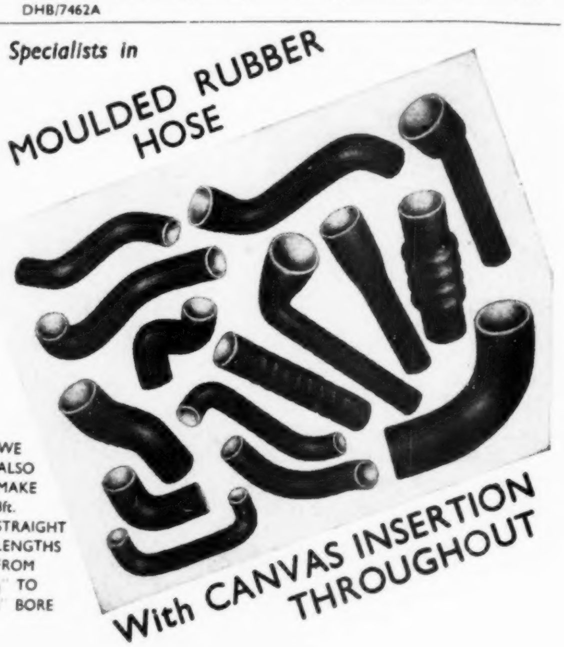

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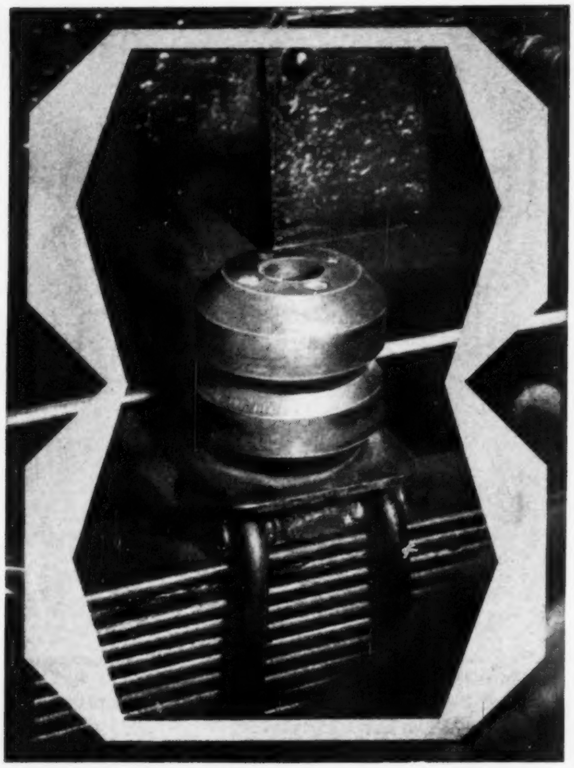

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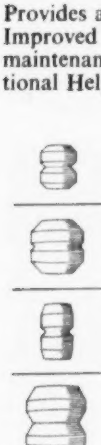

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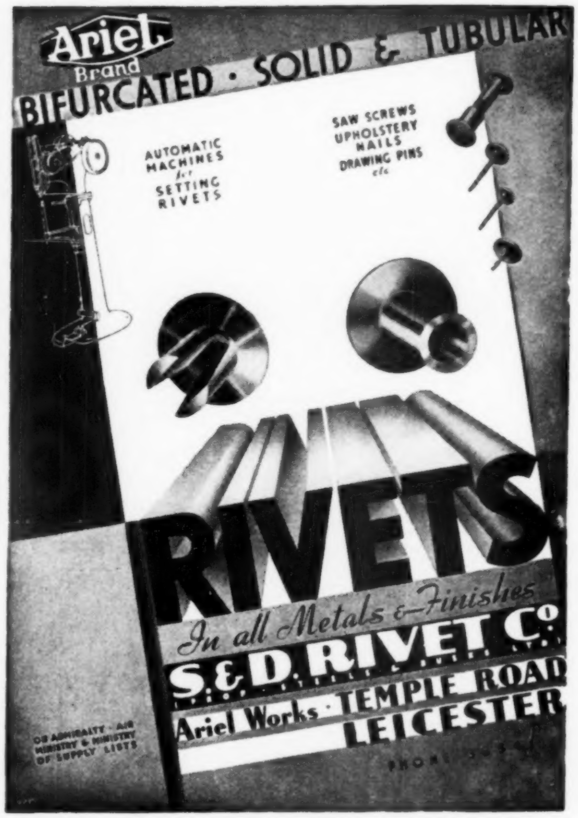

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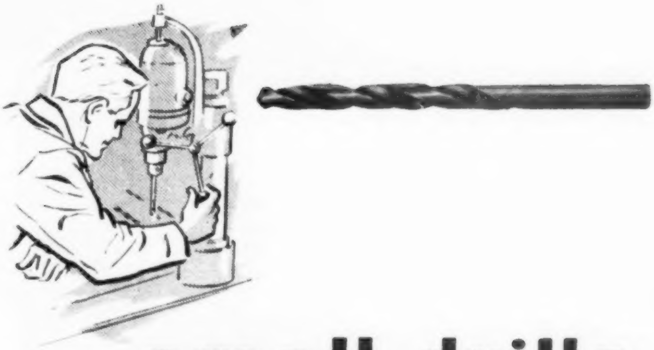

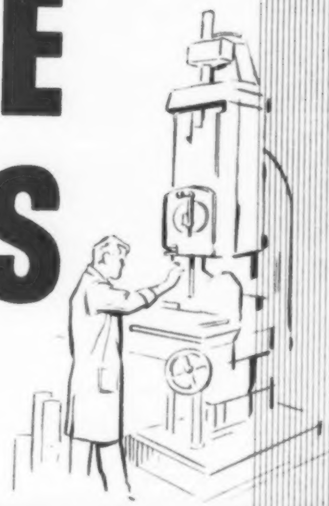

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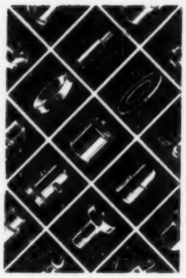

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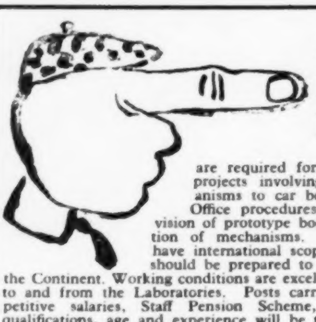

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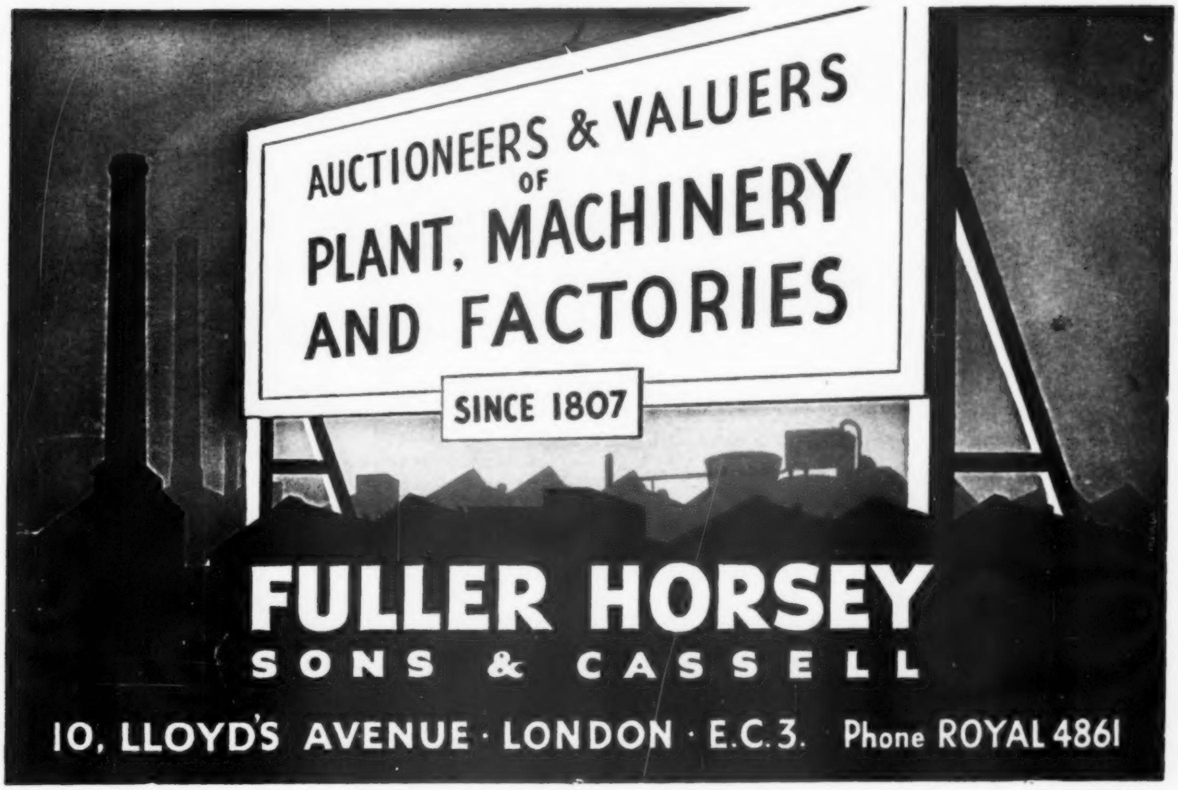

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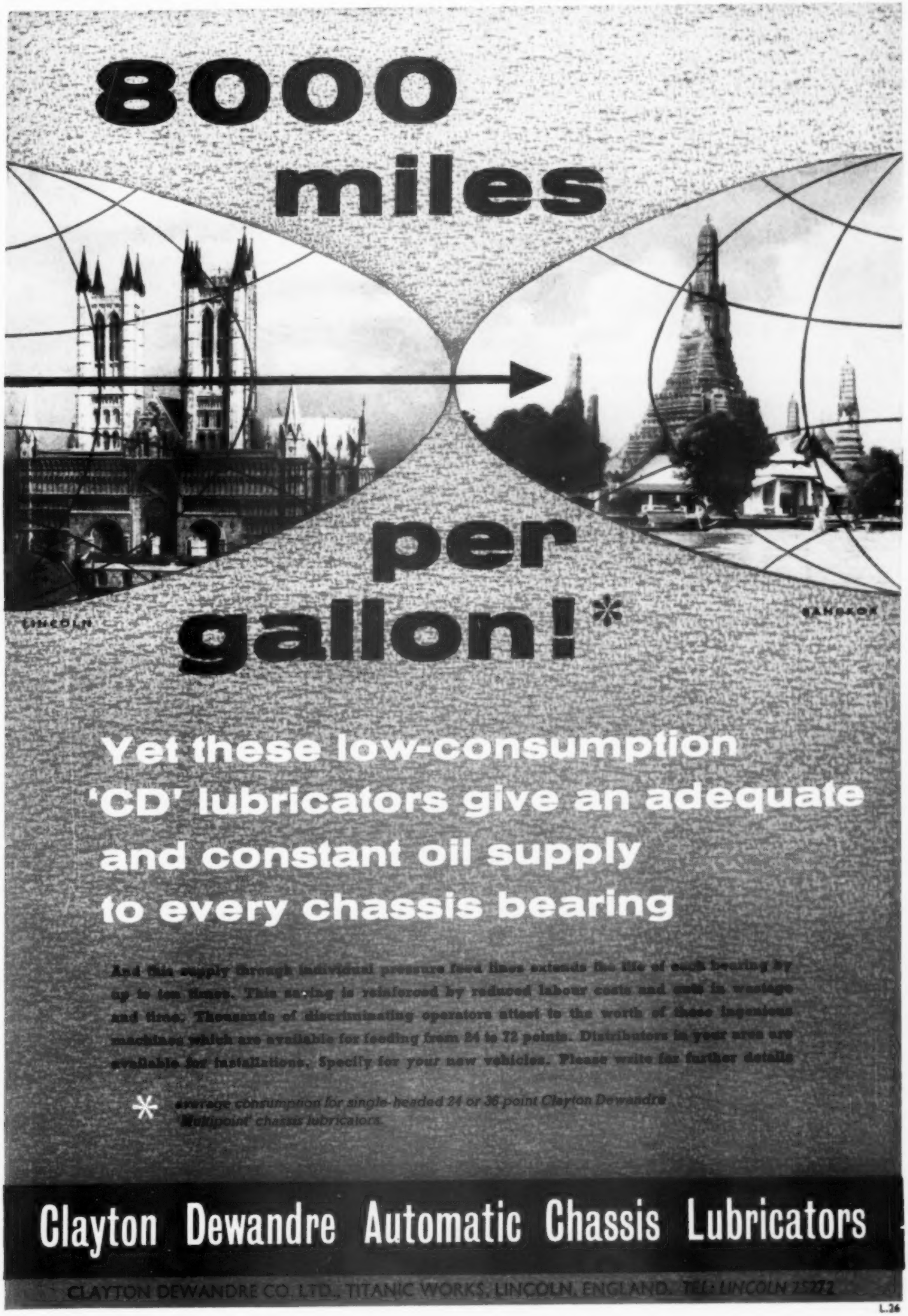
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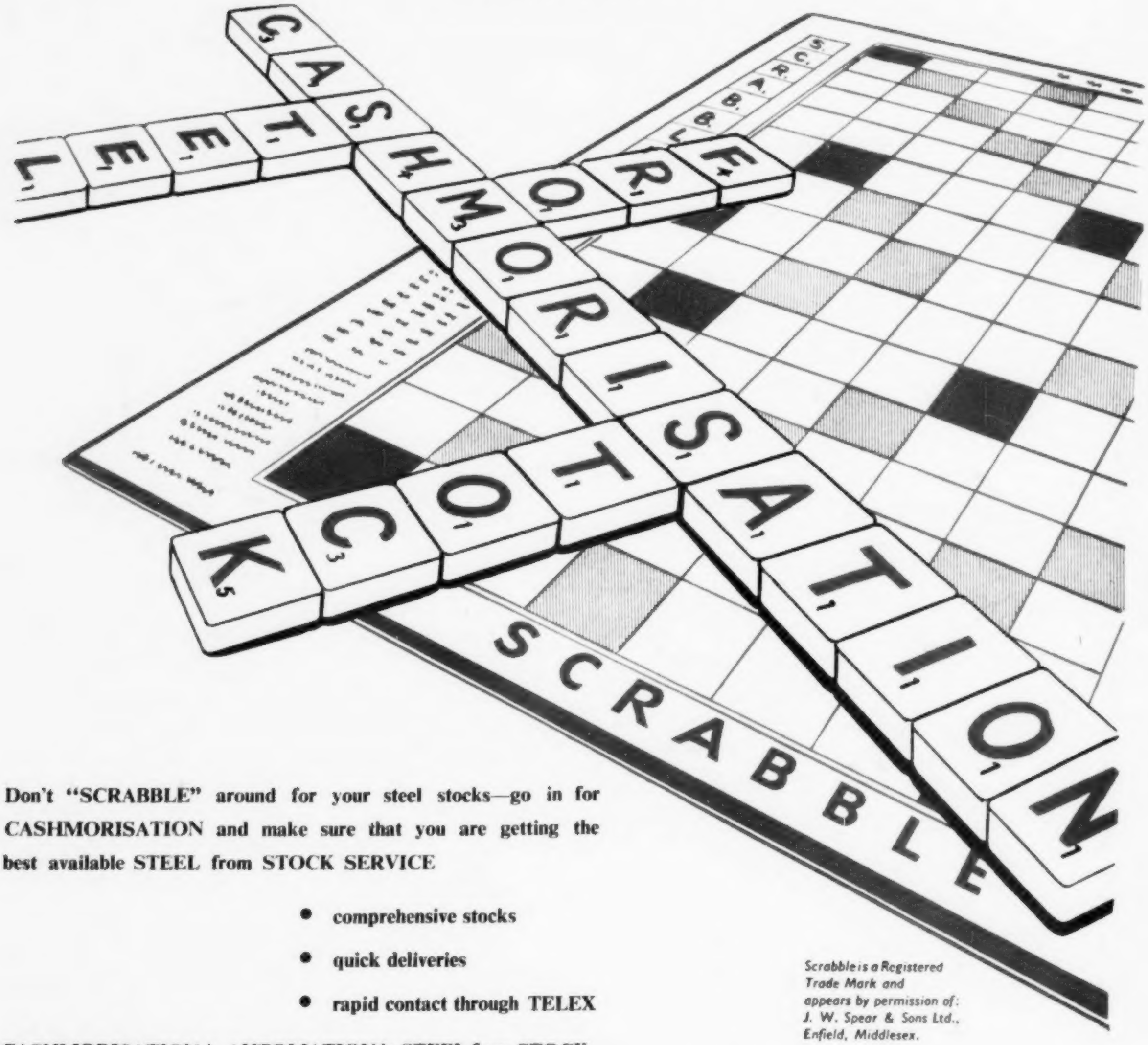

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